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Heating Piping & Air Conditioning

DESIGN
INSTALLATION
OPERATION
MAINTENANCE

COMMERCIAL
INDUSTRIAL
AND LARGE
BUILDINGS

FEBRUARY ★ 1942

e 119—Journal Section of the American Society of Heating and Ventilating Engineers

An unusual application of air conditioning described this month is the cooling of two cabs for cranes handling ingots in a steel mill

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HIT NEW PRODUCTION PEAKS!

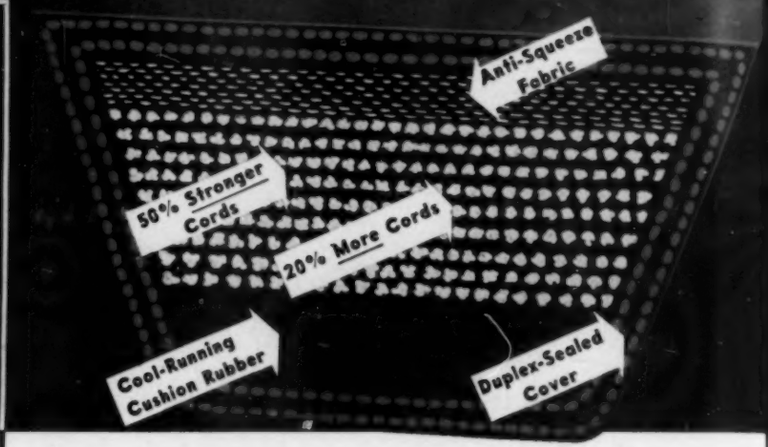
WITH NEW TEXROPE* SUPER-7

1 50% STRONGER CORDS! Sensational new Flexon process makes cords 50% stronger than even those in former great Texrope belts.

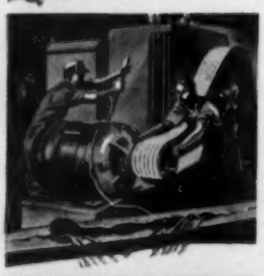
2 20% MORE CORDS! Count them! You get 20% more of these tough, extra pulling power cords in every Texrope Super-7.

3 INCREASED LIFE! Cords float on amazing cool-running cushion rubber that absorbs shock . . . actually increases belt life phenomenally.

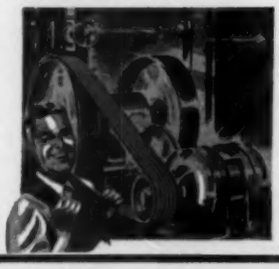
4 EXTRA PROTECTION! Exclusive duplex-sealed cover protects pulling cords against dirt, grit, moisture, and other adverse elements.



LOOK OUT FOR DANGER in belts with stiff, unpliable cords! Such belts *look* strong... but actually they buckle over sheaves, build up excessive heat that attacks pulling cords. Result: Belt failure far sooner than you expect it.



WARNING SIGN! Sagging, slipping belts mean too much stretch not backed up by balanced cord strength. Motor must be pulled back often to take up slack. Result: When stretch limit is reached belt breaks under tension.



SUPER-7 TEXROPE BELTS are made by amazing Flexon process that *combines* flexibility with low stretch... great strength. Cords float on cool-running, shock absorbing cushion rubber. Result: *true* strength . . . *true* pulling power . . . *true* endurance.

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That's why Texrope Super-7 V-belts have a double-barreled value today. For here's a line of V-belts that has been designed and built to give you increased pulling strength and drive efficiency...plus longer belt life under the toughest war-time operating conditions.

In every way these belts are better—

You'll find they have 50% stronger cords! 20% more cords! A new rubber cushion section that effectively absorbs shock! Full protection against heat, dirt, moisture, and other adverse factors that quickly shorten the lives of ordinary belts.

You'll find, too, they give smooth, silent, slipless, high efficiency performance—in "matched" sets, and in sizes and types to meet your every requirement.

Whatever your war-rushed power transmission needs, don't be satisfied with anything but the best—*and that best is Allis-Chalmers Texrope Super-7—the V-belt for Victory!*

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Available in All Sizes
½ hp to 2000 hp

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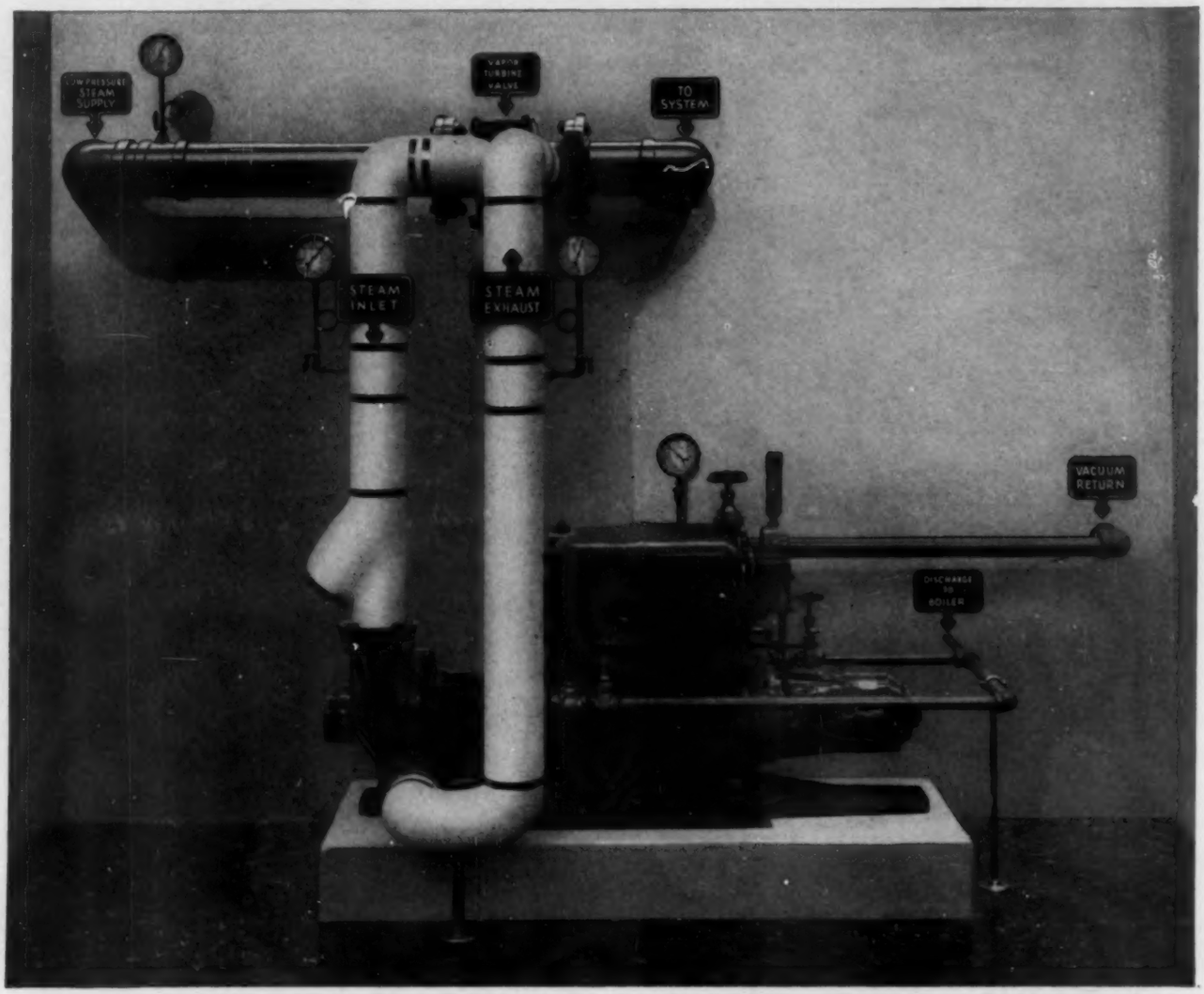
Interrupted Electric Service will not stop this NASH Heating Pump!



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

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Double Duct Air Conditioning System Meets Individual Room Requirements

James N. Livermore, Engineer, Drafting and Surveying Bureau of Detroit Edison Co., Describes Air Conditioning Scheme for Ten Story General Offices Building

THREE years ago the general offices of The Detroit Edison Co. had no central air conditioning system but had instead a collection of cooling cabinets of various ages and types scattered throughout its 10 stories in locations where the need for summer cooling was most pressing. In the few short years of their use these units had grown to 37 in number and were already becoming quite a maintenance problem. It took two men eight hours to overhaul and correct the chronic ills of some of the older types. However, had it stopped there, the servicing of these room coolers might have been tolerable, but it was becoming a certainty that each succeeding summer more of them would be installed. Continuing in this manner until the ultimate was reached might have resulted in the installation of some 420 cabinets by the time every office was cared for. This prospect appeared, in the light of experience already gained, to be a greater servicing job than the building operating force should be called upon to undertake.

At about this same time the company's service building was completed; its air conditioning and design have been described in HPAC for October, 1937, and April, 1938. This building allowed the vacating of considerable space in the general offices and called for an extensive relocation of company departments with partition changes on every floor. It was therefore an opportune time to install an air conditioning system that would take care of the entire building while these alterations were taking place. The decision to do this was made and the work of design and erection was put in the hands of the company's construction bureau.

First Step

The first step toward completely air conditioning the general offices building had already been taken when a 200 ton "Freon" refrigerating plant was installed only a short time before. This plant supplied chilled water to a number of the units already mentioned as well as to two independent duct systems in the building designed for summer cooling only. The chilled water piping, however, had been extended the full height of the building in size adequate to supply any unit or unit system requirements that might occur on any floor in the future.

Zoning

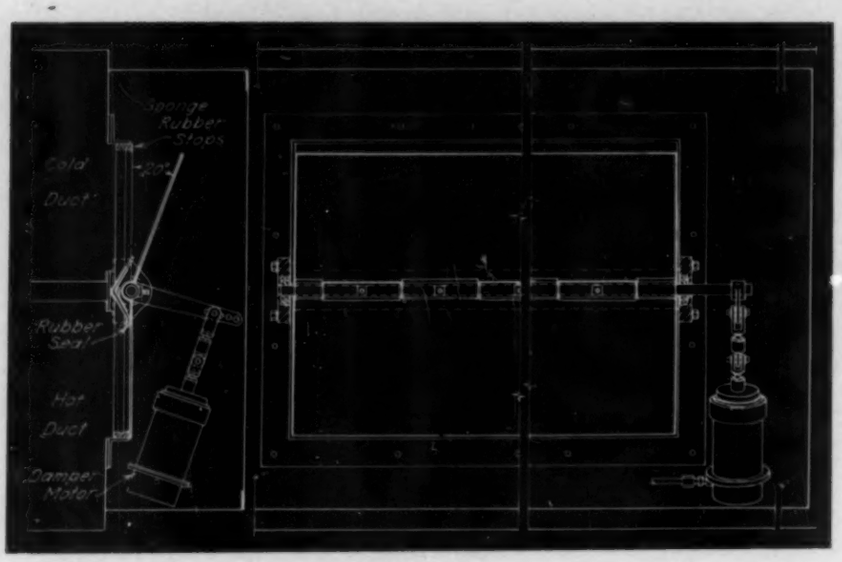
In the earlier studies of this project zoning as a means of temperature control was given serious consideration. Some of the company's previous experiences with zoning, however, had not proved altogether satisfactory. With a group of completely separated offices all receiving air of the same temperature, the occupants of one office often made entirely different use of lights and awnings than their neighbors. Although the volume dampers to compensate for these differences could be readily manipulated by the occupants, they were either not operated at all or were not used until the rooms were already uncomfortable. The excellent performance of the double duct system in the new service building also had great influence on the type of system decided upon in this case. In the service building, the practice of providing separate temperature control for every totally enclosed space has been rigidly adhered to, and the ability to heat one area while another is being cooled has been of tremendous value.

Inability to do this with one of the company's zoned systems made operation very awkward in moderate to cool weather. The occupants were anxious to keep their windows closed at all times to avoid the dirt nuisance and thus created a demand for cooling when open windows would suffice. This created a situation in which the sunny exposures

General offices of The Detroit Edison Co. Maintenance of some of the older individual cooling cabinets used at various points throughout the building had become a problem—and there was certainty of further need for air conditioning—so a central air conditioning system for the entire 10 story structure was installed

---

DOUBLE DUCTS, one carrying warm air and the other cold air, feature the central plant air conditioning system for the general offices of The Detroit Edison Co. This allows individual control in each office enclosure. A constant quantity of air is delivered to each room, its temperature being varied by mixing air from the cold and warm supply ducts as called for by the heating or cooling requirements of the room. . . . A detailed description, such as this one by Mr. Livermore, is not only valuable because of the information included which can be applied to other jobs; it is also worthwhile because it indicates the problems the designer, installer and user of air conditioning face. . . . Based on experience with other air conditioning installations, a number of requirements and objectives were established by Detroit Edison engineers when this job was under consideration. How these requirements and objectives were met in this specific case are covered thoroughly.



SECTION **Detail of mixing damper** ELEVATION

of the building would call for cooling from the duct system while the shady exposures still needed a small amount of heat from the radiators. At such times occupants of rooms requiring heat were annoyed by the air coming from the duct system even though it was being delivered at room temperature.

Double Duct System

Supported by these experiences, it was decided that the general offices would best be served by a double duct system. This means that every office enclosure must have a separate temperature control and that the central system should be capable of heating as well as cooling. This type of system, to operate successfully, must also deliver a constant quantity of air to each room. The temperature varies, however, by mixing air from the warm and cold supply ducts as called for by the heating or cooling requirements of the room.

There were two features of the building which lent themselves well to the layout of a centralized duct system. One was the existence of an easily vacated space originally planned as a shaft for an additional bank of passenger elevators. The idea of installing these elevators had long been abandoned and the space would serve admirably as a location for supply shafts. The second feature was the existence of an exhaust system which drew air from the corridors. It was found that this system could easily be rearranged for recirculation to a central fan room provided the fan room was situated at the top of the building. These two considerations, therefore, fixed all the air handling equipment at the roof where it could be housed by enlarging the existing penthouse.

The general scheme on which the design of the system was based consists, briefly, of two practically independent fan systems, one carrying cold air and the other warm air through parallel vertical main shafts and parallel corridor trunk ducts in such a way that warm and cold air can be supplied simultaneously at every room in the building. The selection and arrangement of penthouse equipment were so dependent on the method of distribution and temperature control of rooms that it will be well to discuss this first.

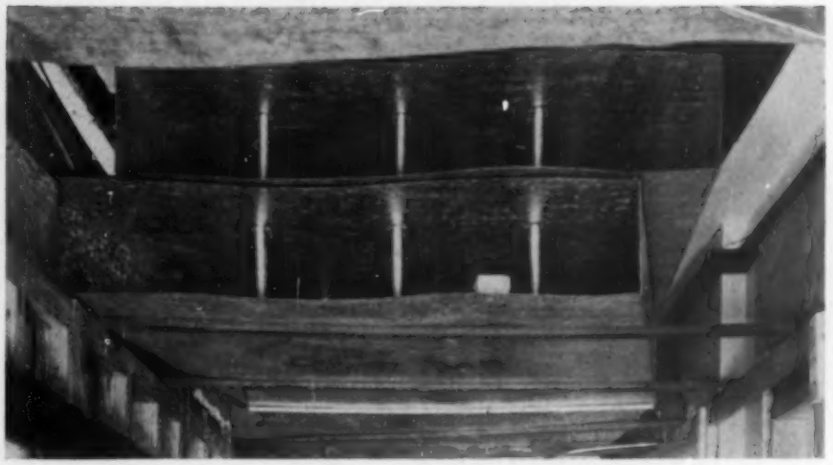
Duct Construction

The main warm and cold supply shafts are adjacent and are lined with non-combustible insulation 1 in. thick. At the takeoff point at each floor there are fire dampers which are held open by compressed air in their actuating motors and which close when the air is released. They may be closed either by the engineer turning an air switch in the basement or by a fusible vent plug which will release the compressed air in case of excessive temperature.

From the shaft takeoffs, cold and warm ducts, separated by insulation, extend the full length of the corridors. Corridor headroom before the ducts were installed was about 10 ft, 6 in., and with the new finished ceiling there is 8 ft, 3 in. headroom. In order to maintain this headroom, trunk ducts necessarily are very wide at the largest sections and presented quite a problem structurally. External stiffeners for duct widths as great as 7 ft would have required more headroom than could be allowed and also would have been very difficult to apply in the working space available. For these reasons, the use of internal streamlined struts was adopted. The lower duct was made to support the upper one by placing the struts one above the other. The load on the struts was then transferred to 2 in., 2.44 lb channels which span between the corridor partitions and are on 24 in. centers. This obviated breaking

Headroom in the corridors before the ducts were installed was about 10 ft, 6 in. and after duct installation it is 8 ft, 3 in., as indicated by these two views. In order to maintain this headroom, ducts are quite wide, presenting a problem

External stiffeners for duct widths as great as those needed would have required too much headroom, so internal streamlined struts were used. The lower duct supports the other one because the struts are one above the other. The load on the struts is carried by channels



through the old ceiling and anchoring hangers to the floor slab above. An economy as well as an additional saving in headroom was realized by using these duct supporting channels as the base for the new corridor ceiling construction. This consists of light ¾ in. channels tack welded at right angles to the 2 in. channels and to which is clipped gypsum lath with a finish of fibrous acoustic tile.

It will be noted here that this arrangement of trunk ducts calls for duct sections that would ordinarily be avoided due to their poor proportions. However, it was possible to size them on a constant pressure loss basis of 0.09 in. per 100 ft and the total supply pressure at the fan outlet is 0.60 in. water gage, which is not judged to be excessive.

Mixing Dampers

At each point of takeoff from the corridors to an office, a mixing damper was mounted with its frame sealed to both trunk ducts. Thus, both cold and warm air can be admitted through the damper and the proportions of each controlled by the position of the damper blades. A pneumatic piston type motor (controlled by a pneumatic room thermostat) is used to position each damper. Previous experience with mixing dampers indicated the importance of design in such devices in order to obtain good temperature control. The most outstanding requirements of them are:

1) Straightline relation between damper motor travel and flow through damper opening.

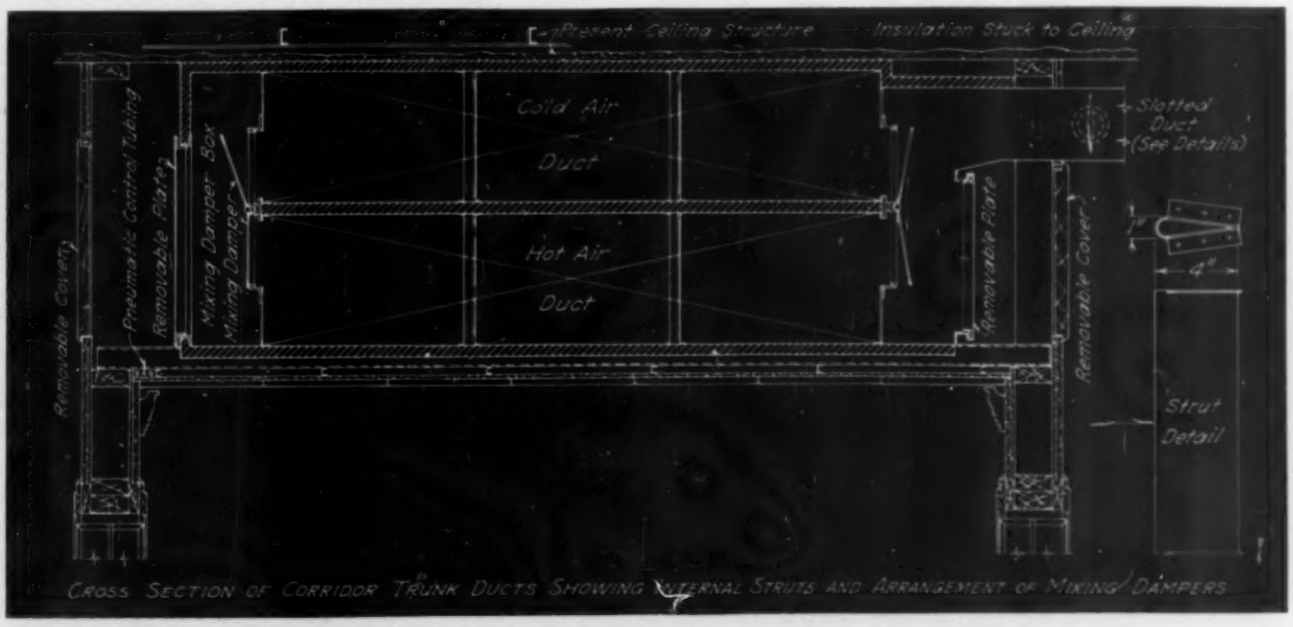
2) Tight shutoff of flow at the end of damper travel.

3) Ease of damper movement and absolute freedom from sticking or "jump."

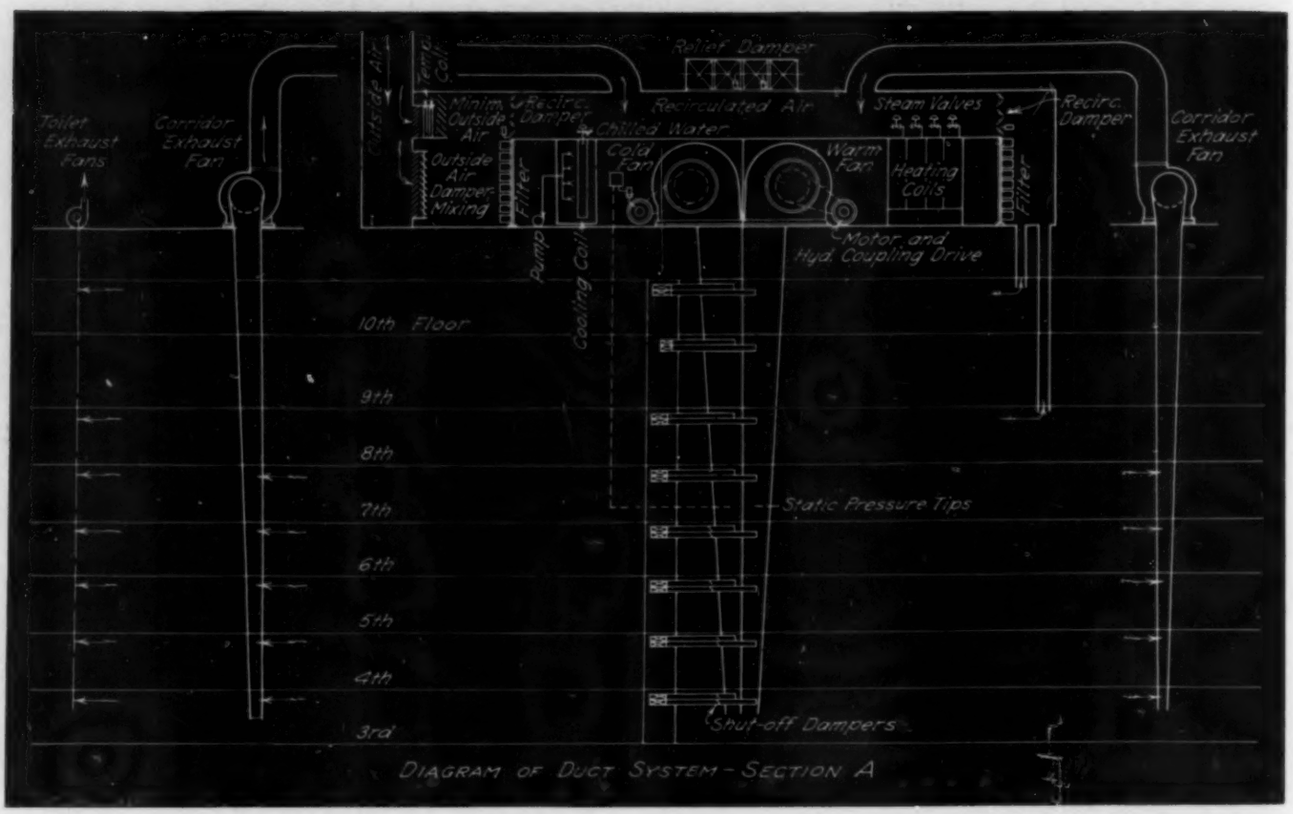
The damper mechanism reproduced here in detail illustrates what was done to meet the above requirements. It will be noted that either damper blade moves only about 20 deg from full open to shut. Over this range a good relation between flow and damper travel is obtained when the static pressure in the trunk duct is held constant. The damper blades were made of No. 14 ga stretcher-level plate and special care was taken to keep them true and to avoid springing the frames during installation. Thus completed, the blades seat tightly on the sponge rubber stops at the end of travel. These dampers were specially made in a machine shop.

Damper Control

In order to obtain good positioning of the damper, a low sensitivity system of control is used; that is, the damper motor spring was specified to be stiff enough that a large change in thermostat branch line pressure is required to move the damper through its stroke. The sensitivity of the thermostat was adjusted so that in a room temperature rise of 1½ F, its output pressure to the motor rises 7½ psi, which is enough to move the damper



CROSS SECTION OF CORRIDOR TRUNK DUCTS SHOWING INTERNAL STRUTS AND ARRANGEMENT OF MIXING DAMPERS



from one extreme to the other. This arrangement makes a maximum of power available to change and to hold the damper blades rigidly in floating positions.

Generally the thermostats are mounted on corridor partitions. In order to improve the responsiveness of the thermostat a 1 in. hole was cut through its back plate and this air passage continued through the partition. Since the offices are under a slight pressure, returning their supply air through door grilles to the corridor, a similar flow is obtained parallel to this circuit over the thermostat sensitive element to the corridor. This practice was followed generally throughout the building.

Each mixing damper is enclosed in a takeoff box which serves to mix the two temperatures of air and supply the duct leading to the room diffuser.

Air Diffusion

The company, in a number of previous installations of air conditioning, has not succeeded in using side wall diffusers, however carefully selected, without having drafts in some portion of the room. This criticism applies also to any other type of diffuser which supplies all air being delivered from one central point. Consequently, the room diffusers used here are of two types, neither of them conventional.

In many cases perforated metal acoustic ceilings were in use, and when they were available they were suspended about 8 in. below the plaster ceiling and the supply air diffused above them. The acoustic pads which are used to back up the perforated metal were removed in a uniform pattern so that about 15 per cent of the ceiling area is open to air flow. This method has proved eminently satisfactory.

Where this type of ceiling was not used, a slotted duct diffuser was installed. This consists of a finished duct with flat soldered seams carried tight against the ceiling and extending the full way from the corridor partition to the outside wall. The diffuser when finished presents an appearance not unlike a furred beam. The inner parts of the diffuser are illustrated herewith in some detail. The diffuser consists of a main flow chamber and a series of outer expansion chambers along the lower corners. Slots in the inner wall feed through dampered openings and through folds of expanded metal to corresponding slots in the outer wall of the duct. The expanded metal serves to break up the jet from the inner slot and air flows through the outer slot at velocities of the order of 200 fpm or less. It will be noted that since there is a damper on each inner slot, flow over the length of the diffuser is controllable and adjustments can be made to eliminate "piling up" at the duct end, and any spotty conditions that may occur. This diffuser was developed after considerable experimentation, in an effort to obtain a truly low velocity diffuser which would not give bad side wall drafts.

Fan Room

The nature and arrangement of equipment in the fan room can now be more clearly understood. The corridor exhaust system was altered to withdraw air only from the third to the seventh floors and to discharge into the return plenum of the fan room or outdoors. This plenum was arranged to maintain sufficient suction from the supply fans to draw return air from the eighth and 10th floors through a separate passage. Return from the ninth floor was omitted since it houses mainly an

assembly room and cafeteria in which separate cooling systems were installed. Makeup outdoor air is also admitted into the return plenum where all the air handled is drawn through cotton bat filters and thence through either cooling coils or heating coils in parallel paths to the warm and cold supply fans.

Under varying load conditions, a majority of the room mixing dampers will demand more air from one supply shaft than the other, and thus the relation between the output of the warm and cold fans must constantly vary. If the mixing dampers are to deliver unvarying amounts of air the static pressures on the cold and warm duct systems must be constant and equal regardless of flow. This was accomplished by automatic control of the speed of each supply fan. Variable speed hydraulic couplings are used in conjunction with v belt drives between fans and couplings. The couplings are driven by constant speed squirrel cage motors, and the output speed of each is varied by positioning the coupling oil scoop. Pneumatic type static pressure recorder-controllers are used, with their pressure tips at about mid-height of the supply shaft.

Flooding Nozzles

It was desired to use as low an air temperature in the duct system as practical in order to keep the duct sizes at a minimum. Extended surface cooling coils were preferred, but since they do not ordinarily reduce the leaving air to a saturated condition, they would in this case remove an excess of latent heat in order to remove the required amount of sensible heat. This could be corrected by adding several more rows of coil depth with the attendant additional fan static pressure load, or by installing flooding nozzles with a pump to spray recirculated water on the coil face. This latter arrangement was chosen and was calculated to reduce the coil load by 26.5 tons of refrigeration in excessive latent heat removal. Spraying the coil has also proved to be of some value for evaporative cooling when chilled water is not being circulated through the coil.

The heat for the warm duct system is provided by sections of finned coils installed at the intake side of the warm fan, and supplied with 5 psi steam.

Temperature Control

The temperature of the cold shaft is controlled thermostatically by throttling the flow of chilled water through the cooling coils, and the temperature of the warm shaft is controlled similarly by throttling steam to the heating coils. The thermostats proper are mounted on the main control panel in the basement where they can be easily regulated by the operating engineer. The sensitive elements are in the warm and cold shafts and connect to the thermostats through long runs of capillary tubing. Under certain outdoor weather conditions the control of the cold shaft can be shifted from the chilled water throttling valve, to perform the same function by operating dampers in the penthouse which proportion quantities of recirculated air and cold outdoor air. Air switches on the main panel also permit the engineer to recirculate air or not, depending upon the outdoor wet bulb, and to select the number of sections of heating coil to be in use.

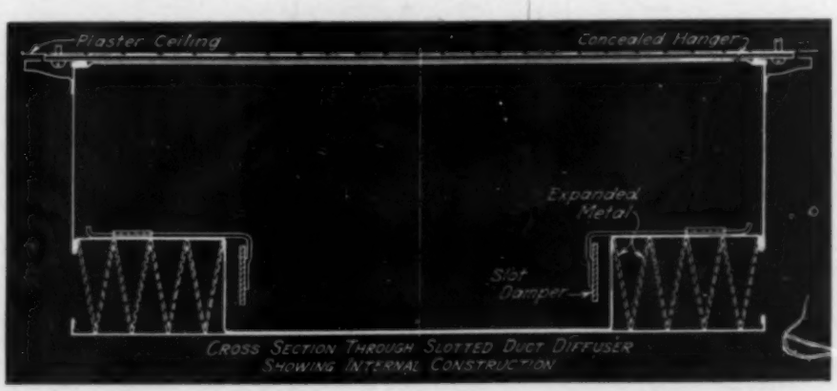
For winter humidification, a small spray type humidifier with steam heater was installed in parallel with the heating and cooling coils and part of the air is diverted through it. This operates intermittently in response to a hair element humidistat mounted in the corridor air stream.

In addition to controls and air switches already mentioned the main control board in the basement also has bull's eye lights showing what equipment is in operation at the roof and stop buttons for all motors. The board also carries remote reading instruments for all pressures and temperatures needed to operate the equipment properly. Therefore, it is unnecessary for the engineer to go near the penthouse except for routine inspection and maintenance.

Refrigeration

In order to obtain refrigerating capacity enough to operate this system, the present 200 ton machine in the basement is used and by extending two chilled water lines through the subway between the general offices and the nearby service building, the surplus capacity in the service building was made available. During the first season of operation, this combination was adequate except for the hottest days, and the amount of additional refrigerating capacity to be installed will be determined from operating data and the need for standby capacity.

During the one full heating and cooling season that this system has operated, the extremes resorted to for good control and air distribution appear to have been thoroughly justified.



Typhoid-paratyphoid vaccine is made at the Army Medical School in the largest laboratory of its kind in the world. The planting and harvesting rooms, made of monel metal and glass, are sterilized with live steam. During the various steps of vaccine making, the chambers are sealed, and conditioned air, which is thoroughly sterilized by washing and passing through ultraviolet light, is pumped in under positive pressure. All of the typhoid-paratyphoid vaccine required for use by the army, the navy and many other federal departments is made at this laboratory, which at peak production can provide one and a half million doses of the vaccine every week. During the last fiscal year the output was increased over 800 per cent.

BUYING AND CHECKING FUEL

New Procedure Used by Building Management Firm Provides for Guaranteed Heat Content and Definite Penalties for Poor Performance

By R. E. Hieronymus

COMMERCIAL laboratory analyses of coal from the bin are provided for in the coal contract form we use, and which was reproduced in my article on buying and checking fuel published in the January HPAC. This contract form provides for guaranteed Btu values of the coal, a definite number of heat units for one cent and testing of the coal on an "as received" basis; it includes definite penalties for poor performance of the contract terms.

Coal Testing

For those who are not familiar with the making of coal analyses, the commercial testing procedure is briefly described:

A small quantity of the coal is taken from the bin, using a standard method designed to assure that the sample is really representative of the supply on hand. If the laboratory is situated in the same town, it is well to have them send a man to the plant or building to take the sample.

At the laboratory the chemist will analyze part of the sample, again using one of the several standard procedures for determining the characteristics of the fuel. Using highly sensitive scales, he will weigh a portion of the sample (after pulverizing it) just as it came to the laboratory. Then he will dry it slowly to drive off the moisture, weighing it again thereafter to determine the percentage of moisture it contained.

Then he will burn a measured quantity in a little specially constructed autoclave, called a bomb, measuring (in Btu's) the amount of heat which it gives off. This apparatus is known as a calorimeter. He will then perform other tests to determine the amounts of volatile matter, fixed carbon, sulphur and ash, reducing them to percentages.

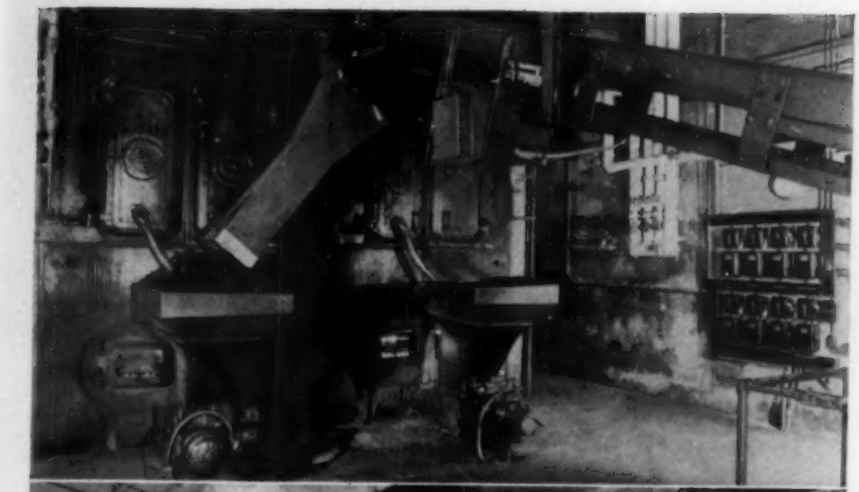
FUEL purchases represent a major item in the average plant or building — in office buildings, for example, fuel for heating represents the largest amount paid for any commodity, after electricity. It is therefore important to economical and efficient operation and management that the most suitable fuel be used — particularly is this true now because of the need for conservation of everything possible for war production. . . . The right fuel for a plant or building may change from time to time with changes in cost of coal, oil and gas and with other factors, as pointed out by Mr. Hieronymus in the January HPAC. The basic figure to be used in selecting a fuel, or contemplating a change, is the Btu's which may be bought for one cent. . . . Because routine analyses of coal used in 11 buildings during one year showed 88 per cent to be of a lower Btu content than represented and only 12 per cent higher than represented, Mr. Hieronymus now uses a contract form which provides for guaranteed heating value of coal, with provisions for testing and penalties for poor performance. The method was described, and a sample contract form was reproduced, in January. This month, coal testing is discussed and the meaning of an analysis is explained. . . . The author, a member of HPAC's board of consulting & contributing editors, is the general superintendent of Aldis & Co., large Chicago building management firm

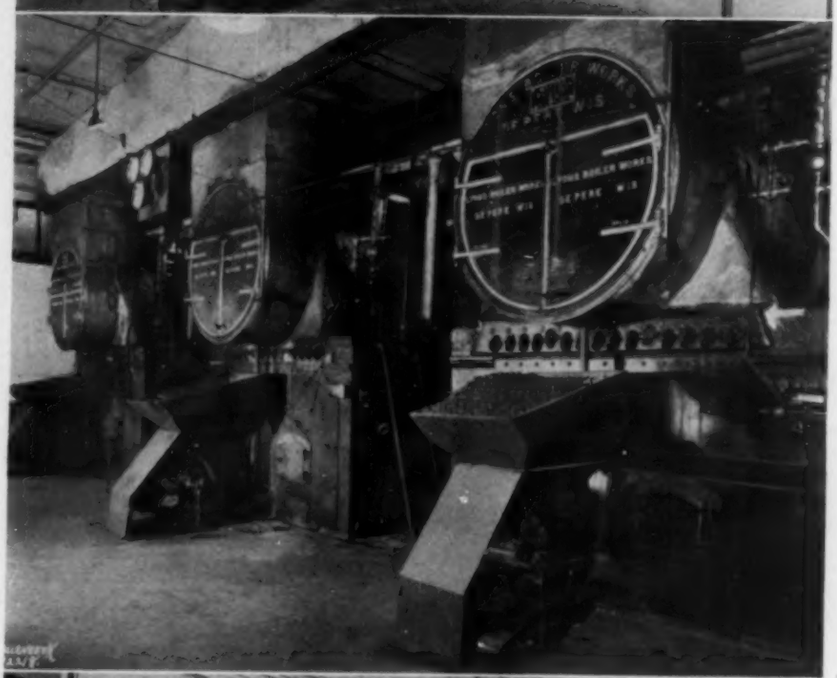
Proximate Analysis and Ultimate Analysis—Two types of analyses are possible, the proximate analysis, which is the less expensive and which fully covers the needs of the present discussion, and the ultimate analysis which is more exhaustive, more expensive and need not be considered here. It is not necessary to tell the laboratory what grade of fuel you are submitting for analysis —partly for the reason that you can't be sure yourself that it is the coal you thought you were receiving.

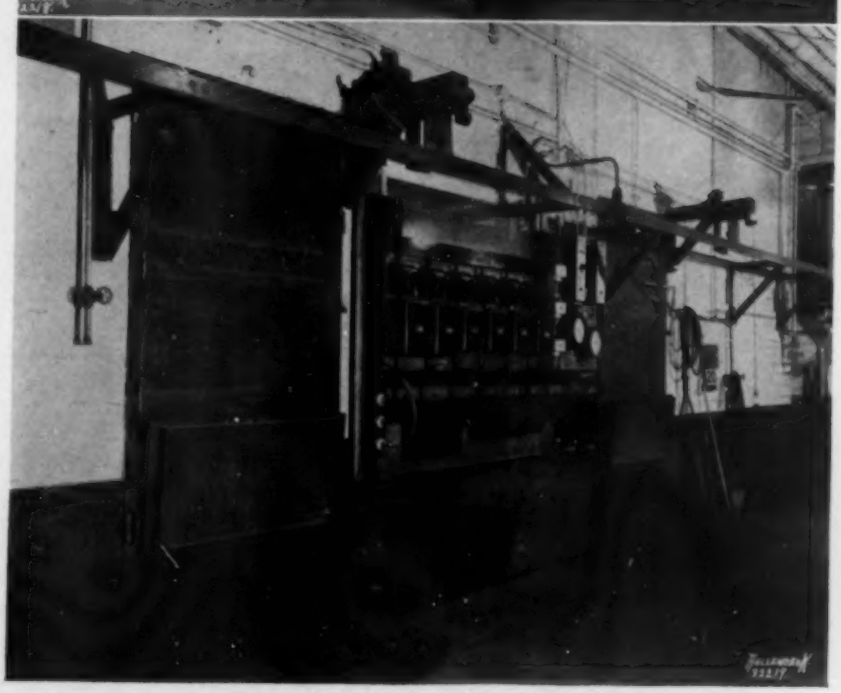
As Received Basis—The laboratory will furnish the analysis in two parts. One, called the as received basis, is the analysis of the coal just as it comes from your bin, including a determination of the amount of moisture (both inherent and extraneous). The inherent moisture in coal is a part of its intimate chemical structure, and the extraneous moisture is present with the coal in a physical rather than a chemical sense. Around the coal mines the extraneous moisture is commonly called surface moisture.

Dry Basis—The other part of the analysis, called the dry basis, shows the characteristics of the coal after all the moisture has been dried out of the sample by means of artificial heat.

While the dry analysis is valuable in the coal industry for some purposes, it is practically of no use to the plant or building operator. The moisture is rather a variable with any particular coal, and since the purchaser buys and uses the coal on the *as received basis* he buys the water right along with the coal—and pays a fancy price for it since he is buying it by the ton at the







ton price of coal. The building (or small plant) purchaser (and user) of coal need not care whether he ever sees a dry analysis—the as received basis is of sole importance.

(It should be mentioned that the laboratory does not duplicate the work of testing when they furnish both analyses—they make a complete analysis and then calculate it in percentages both ways, on the as received basis and on the dry basis.)

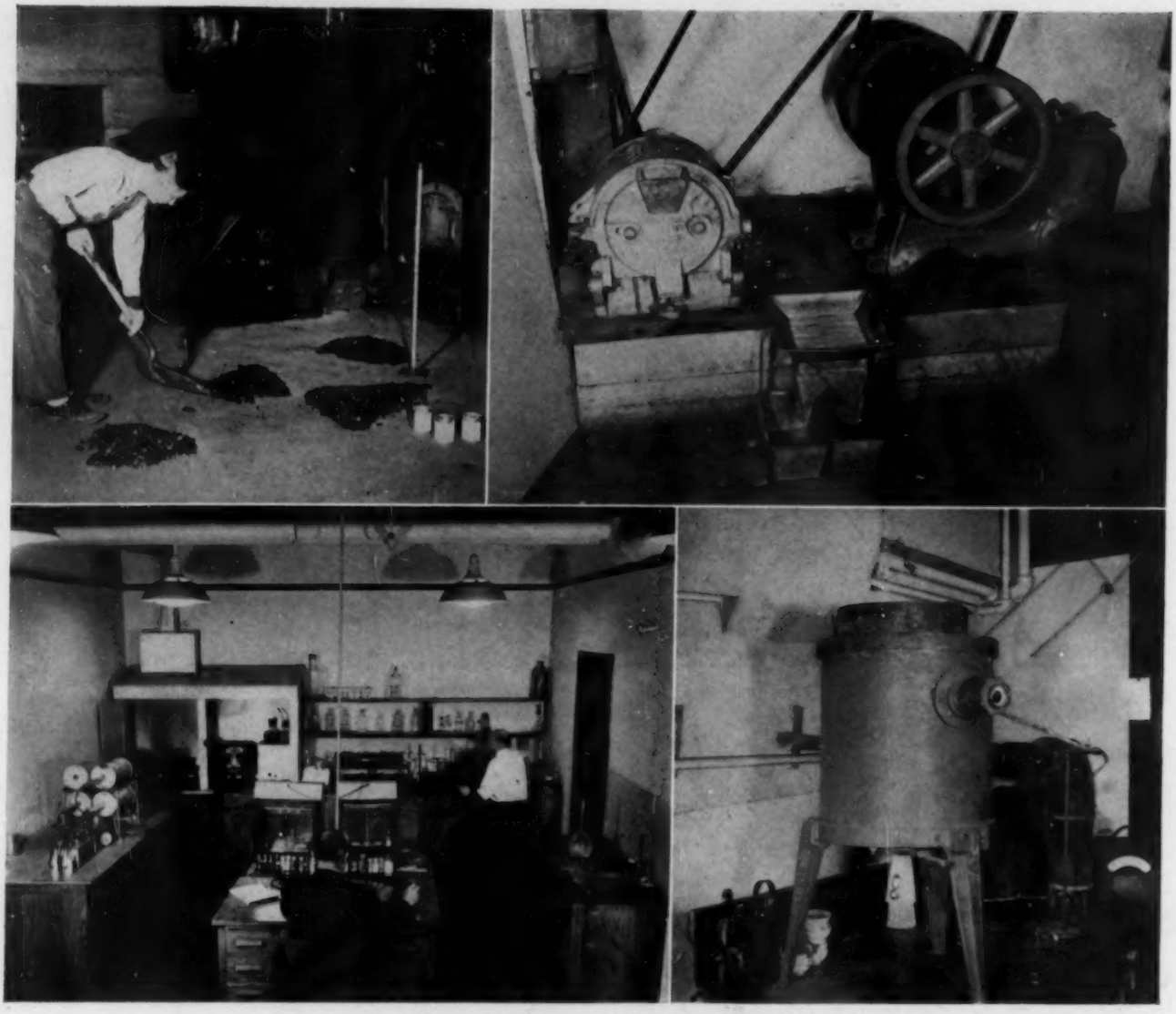
Moisture in Coal

The subject of moisture in coal, and its effects on the efficient burning of the fuel, is a large and intricate one and need only be touched upon briefly here.

The moisture does not take part in the process of combustion, but in the burning of the coal the moisture is changed into steam (requiring about 1000 Btu of heat to change each pound of it) and further superheated; and when this steam goes out the smokestack each pound of it takes with it the 1000 Btu, plus the energy of superheat, which it gives up to the atmosphere when it finally cools and again condenses into water. Thus it is seen that any excess moisture in the coal (beyond that required in some furnaces for orderly combustion) results in the needless loss of part of the otherwise available energy in the fuel. Certain fuels when used with certain equipments require the addition of extra extraneous moisture (by wetting) to cause them to hold together better on the grates during the combustion process. The operator balances the loss-evil of inefficient burning with the alternate loss-evil of high moisture in the coal, and seeks to operate with the most economical low-loss compromise between them.

In general it is probably true, for office building plants, that the coal

Boiler plants in some of the buildings under the author's supervision. . . . *Top*—Two underfeed stokers serving 85 hp HRT boilers, and fed with swing spout coal conveyor direct from coal bin; controls are on panel at right. . . . *Center*—Three underfeed stokers serving three combination watertube and firetube boilers, 150 hp each, operating at 100 psi ga pressure. . . . *Bottom*—Stoker and heating controls mounted on front of coal bunkers opposite the stokers shown above



SOME OF THE STEPS IN ANALYZING COAL

Top left—The final quartering, part of the process of obtaining the sample of coal from the supply in use in the boiler plant. . . . *Top right*—The crusher (right) and grinder (left) used in the laboratory to prepare the sample of coal for analysis. In lower foreground is the riffler, a device for further mixing and reducing the quantity of the sample. . . . *Bottom left*—View of the coal analyzing department in a commercial testing laboratory. The chemist at the bench on the right is putting a coal sample through the bomb calorimeter to determine the heating value (Btu per lb) of the coal. Also visible in this picture are balance scales for weighing, furnace for determination of carbon and hydrogen in the ultimate analysis, muffle furnace for ash determination and other associated apparatus. . . . *Bottom right*—High temperature furnace used for determining the fusion temperature of the ash. (These four pictures courtesy Robert W. Hunt Co.)

will be wet enough, or too wet, as it comes from the dealer. Certainly it is true, in the midwestern districts, that extra moisture is a problem and a loss more frequently than is the lack of it.

A Sample Analysis

It is probably not amiss to examine a routine coal analysis and discuss it as a means of checking the value received from the dealer. The first column of the tabulation shows, in one experience, the coal dealer's analysis of the fuel he had contracted to furnish. The second column shows the analysis found by the laboratory on a sample taken from the bin. The fuel in this case is a well known Indiana coal, used in the 1¼ in. screenings size in

Proximate Analyses (As Received Basis)

	As Represented	As Delivered
Moisture	10.32%	14.08%
Ash	8.05	10.52
Volatile	36.45	36.11
Fixed carbon	45.18	39.29
Total	100.00%	100.00%
Sulphur	2.35%	2.35%
Btu per lb	11,664	10,893
The purchaser may add, if the price of the coal be $4.50 per ton:		
Btu for 1 cent	51,840	48,413

Deficiency: 51,840 — 48,413 = 3427 Btu, or 6.6%.

furnaces equipped with underfeed stokers.

In this case the quality of the fuel, on the contract basis, was deficient by 11,664 — 10,893 = 771 Btu per lb, or 6.6 per cent.

What It Means

On our former basis of fuel purchasing, in order to learn these sad conditions, we had to pay the laboratory for the analysis; for each ton we had to dispose of about 2½ per cent (50 lb) of extra ash; for each ton of the coal burned, with 3.76 per cent extra moisture, we had to turn 75.2 lb of water (3.76 per cent of 2000 lb) into steam in the firebox, and lose the 75,000 Btu (75 lb × 1000 Btu) out our smokestack. And the coal dealer, in the face of the 6.6 per cent deficiency in heat units, merely regretted the poor batch of coal his producer had sent him, and hoped he could get him to do better.

On the new, guaranteed basis we now use, the dealer would find himself paying for the laboratory analysis, which would amount to approximately 80c per ton for 10 tons, and, the delivered value of at least one load being 10,893 ÷ 11,664 × $4.50 per ton, or $4.20 per ton. would grant a credit of $3.00 (10 tons × $0.30) on account of the delinquency.

It has been our practice to overlook deficiencies of 2 or 3 per cent, and within this range we pay the testing bill and ask no adjustment.

Too many buyers select a coal on the basis of a glowing description and attractive price per ton, without ever requiring the dealer to put a Btu price tag on his product and without ever checking the "value" of a single ton of it. It is not sufficient merely to compare the fuel bill with previous years on a calendar month or a degree day basis, for the previous years are more than likely subject to the same discrepancies, and the efficiency of the heating process upstairs may not be comparable.

Testing Frequency

To check the delivered value of your coal along the lines of the foregoing is not a difficult matter and the technical aspects should not prove forbidding. The coal dealer proposes one figure; the laboratory, by analysis, produces another; the buyer compares the two, draws his conclusions, and governs his procedure therefrom.

Assuming that performance of coal contracts follows the general outline specified above, and assuming that the making of test analyses would insure the delivery of a fully represented value, a purchaser could afford to have one laboratory analysis made for each $100 worth of fuel purchased. Though this ratio should apply to relatively small consumers, and need not be maintained for large annual consumption, the point here emphasized is that no plant or building should refrain from check analyses merely because the fuel bill is only a few hundred dollars per year; the dealer still needs occasional scrutiny and the cost of analyses is likely to be self-liquidating in better performance of the contract. The coal dealer should understand that the laboratory will be called to take a sample from the bin and make the analysis at irregular intervals.

The frequency of testing should depend, first, upon the tonnage used in a given plant or building and, next, upon the performance of the coal. The plant engineers should be instructed to notify the office when any consignment of coal seems to vary greatly from the customary standard, and in such cases a laboratory test may be ordered at once.

AIR CONDITIONED BLAST FURNACE

THE benefits obtained by equipping one of the blast furnaces at the Aliquippa Works of the Jones & Laughlin Steel Corp. with a three grain pre-compression chilled water air conditioning system are presented. The furnace has 28½ ft hearth diameter, 90 ft stack height, 30 ft bosh diameter, and 18 tuyeres.

During the first six months' operation with air conditioning (April-September 1941), the average monthly production of iron was 8.22 per cent greater and coke consumption was 3.03 per cent less than the monthly averages for the year 1940. For June-September 1941, iron production was 16.07 per cent greater, and for June-August 1941, coke consumption was 4.26 per cent less than for comparable months in 1940. The furnace operates more smoothly, drives more regularly, and keeps a more constant wind blowing, more uniform heat, and a more uniform burden, resulting in a more uniform temperature and composition of iron.

To confirm these results, conditioned air was transferred from the original furnace to another one of the same size during the first half of September and again put back on the first furnace during the second half of the month. For the first 13 day period, production of the original furnace decreased 14.5 per cent, while that of the second furnace increased 13.9 per cent. During the second half of the month, the first furnace, with conditioned air switched back to it, recovered the production lost, while production of the second furnace decreased correspondingly.

During the first six months' operation, the steam saved on the turboblower in blowing the furnace on air conditioning was equivalent to a saving of 2,900,000 kwhr. This more than offset the operating cost of the air conditioning unit, after taking into account the loss of heat in the air entering the stoves. It is expected that when including the winter months, the average increase in production throughout the year will be at least 8 per cent and the decrease in coke consumption will be about 4 per cent.

Some think that it is necessary to reduce the moisture to one grain to obtain any real benefits, but the results stated above indicate that a greater percentage of the maximum tonnage that can be expected will be produced from a three grain system. The author's opinion is that if the moisture is maintained at much less than three grains, the furnace will have a tendency to hang and work on a higher pressure and will produce a poorer quality of iron.—*Abstract from Metals and Alloys of paper by E. K. Miller published in Blast Furnace and Steel Plant, November, 1941, and in Steel, November 10, 1941.*

Modernized Heating System for Enlarged Factory Keeps the Operating Cost Down

When the plant was enlarged to increase floor space by 86 per cent, the heating system was modernized. The increase in fuel cost for heating has been found to be but 16 per cent

A MANUFACTURER of precision equipment enlarged his plant two years ago to increase the floor space by 86 per cent. At the same time, the old heating system was converted to a one pipe, forced circulation, hot water system, designed for 240 Btu heat emission.

In making the changeover, the original piping in the old plant was salvaged to a large degree, thereby saving the owner a considerable sum. The radiation is varied in character to fit in with the functions of the different departments in the building. For example, there are cast iron radiators and unit heaters in the shop sections, copper convectors in the offices, and indirect heating coils are used to temper the air for the ventilating system.

For accurate and economical control of the heating, the system is divided into seven zones. Forced circulation to the zones is supplied and controlled by seven pumps and 18 flow control valves. One hundred and eight pairs of diverting tee fittings connect the risers to the single main and assure a proper diversion of heated water to the radiators, convectors and ventilating system coils. The unit heaters are connected to a two pipe, reverse return circuit, which was installed as a separate zone.

FLOOR SPACE 86 PER CENT MORE BUT HEATING COST INCREASE IS ONLY 16 PER CENT

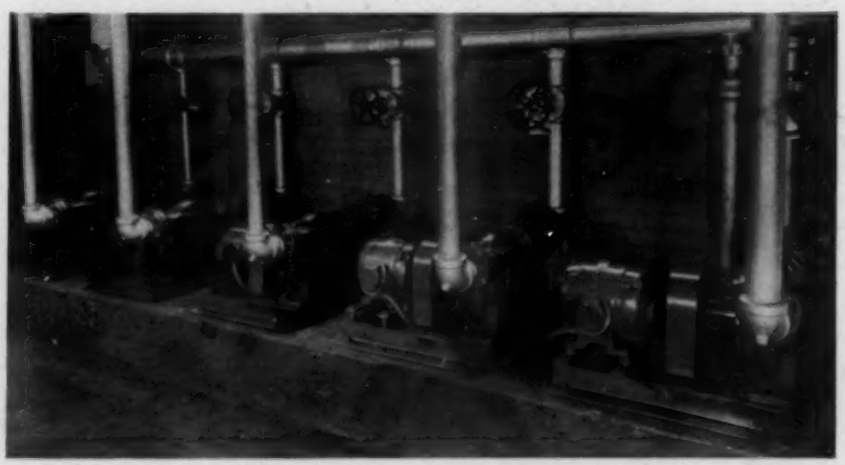
A breakdown of the total 14,515 sq ft EDR shows the following division of radiation types: 3334 sq ft in unit heaters, 7891 sq ft in cast iron radiators, 2090 sq ft in copper convectors and 1200 sq ft in ventilating system coils.

The most interesting feature of this heating installation, from the owner's standpoint at least, is the "before and after" operating cost. The accompanying figures are justification for the complete remodeling job which was done. An analysis of them shows that an increase of 86 per cent in floor space added but 16 per cent to the heating cost.

HEATING COSTS	FLOOR SPACE
Old System	Old Building
Year	
1938 .....$1692.10	20,039 sq ft
1939 ..... 1871.88	
New System	Old and new buildings
1940 .....$2000.81	combined
1941* 2131.21	27,143 sq ft

*This figure is the actual fuel cost up to November 1, 1941, plus an estimated cost of $753.47 for the remaining two months of the year. $753.47 was the actual cost during January and February of 1940.

The forced circulation heating system is divided in seven zones, each of which is served by a pump



CODE FOR PRESSURE PIPING

Alfred Iddles Discusses Requirements of Section 1 on Power Piping Systems

IN the January, 1942, HPAC, outstanding differences between the first and second editions of the American Standards Association's code for pressure piping (B31) were enumerated by Sabin Crocker and detailed explanation was made concerning changes which are common to several sections of the code but which do not appear in either of the generalized sections Nos. 6 or 7 (on fabrication details and materials). The present discussion is devoted to the new requirements for power piping systems which are peculiar to section 1 alone. Items mentioned in the previous article, although applicable to the power section, are not repeated herein.

PIPING CODE—Since the ASA code for pressure piping was first issued as a tentative standard six years ago, there have been many developments and advances in piping practices, as described by Sabin Crocker last month in the first of a group of articles—of which this is the second. The piping code has been thoroughly revised and is being reissued as a full American standard. . . . Section 1 of the code, on power piping systems, is discussed by Mr. Iddles, who is chairman of the section on power piping systems and application engineer with the Babcock & Wilcox Co. He outlines the scope of this section, and gives particular attention to the relationship of the piping code and the ASME boiler construction code; this relationship was clarified by a conference of a joint committee. Pressure-temperature swings and their effect on design, boilerfeed piping, bolting, hydrostatic testing, oil piping and the use of standards are also treated

Scope

According to section 1, power piping systems shall be understood to comprise all steam, water and oil piping (not including oil piping covered by section 3 of this code) within or forming a part of the mentioned plants and to exclude gas, air and refrigerating piping; central and district heating steam or hot water distribution piping away from the plant; building heating piping when the pressure does not exceed 15 psi ga; roof and floor drains; plumbing; sprinkler systems; piping for hydraulic pressure tools or equipment; sewers; and industrial process piping for fluids not mentioned above.

Valves, fittings and piping for boilers, as prescribed in the American Society of Mechanical Engineers' code for power boilers, are within the scope of this piping code, except that provisions of the ASME power boiler code shall govern in so far as they exceed corresponding requirements of this code.

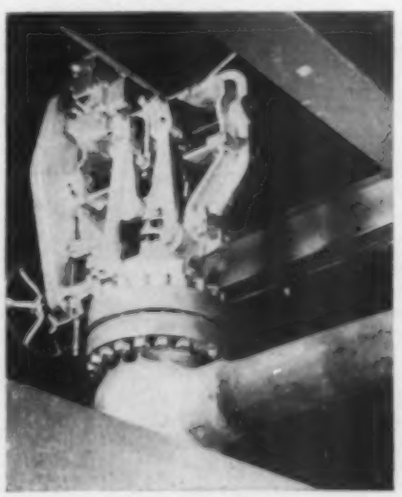
Economizers, heaters, tanks and other pressure vessels are outside the scope of this piping code, but connecting piping shall conform to the requirements specified.

Specific reference to the power section is made under section 3, on oil piping, where it is stated that the rules for plants devoted principally to the processing and handling of oils do not apply to steam piping within the limits of the boiler house or boiler plant areas of refineries, nor for boiler feedwater piping between feed pumps and boilers therein. Such piping must conform to the requirements of section 1. Also, in the case of the district heating section, piping in the steam generating plants is covered by section 1 as well as piping for pressures in excess of 600 psi, or temperatures in excess of 750 F. The scope of the power section has been extended to cover piping operating at temperatures up to 1000 F instead of the former limit of 750 F, but the pressure limit for this section of the code has been retained at 1500 psi.

Agreement with Boiler Code

Particular effort was expended by the sectional committee to reach agreement with the ASME boiler construction code requirements covering piping. The importance of harmonizing these requirements has been emphasized in the recent past by a growing practice to minimize the number of valves in the connection between a boiler and a turbine and to place these valves at considerable distances from the boiler.

The ASME boiler construction code rules for power boilers state that the rules contained therein "apply to the boiler proper and pipe connections up to and including the valve or valves as required by this (boiler) code." In many cases, the boiler code requires a second valve on feedwater and superheated steam piping, and frequently this is placed remote from the boiler. Hence the intervening piping falls within the scope of the boiler code, although essentially it is the type of piping which section 1 of the piping code is intended to cover. Figs. 1 and 2



Stop and check valve on superheater outlet of high pressure boiler. This piping comes within the scope of the ASME boiler construction code and the American standard code for pressure piping

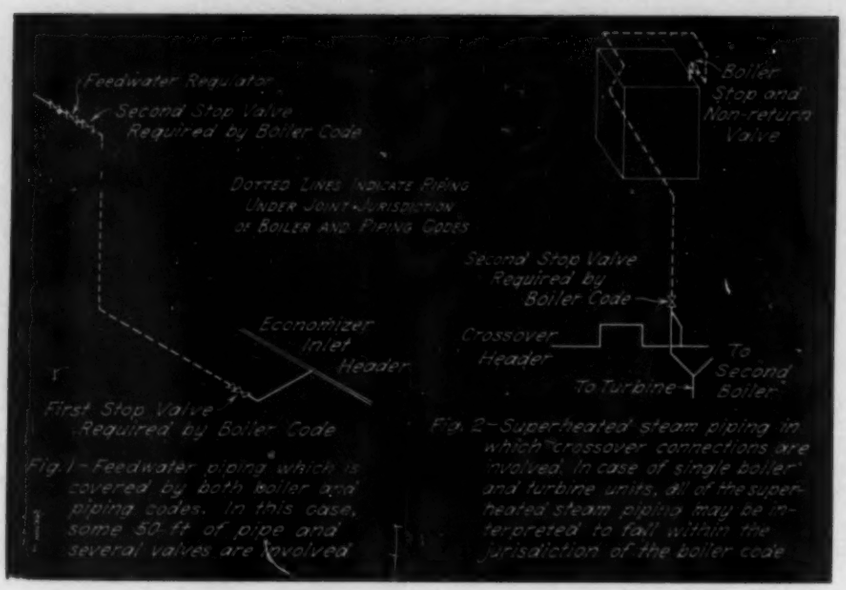

Feedwater Regulator
Second Stop Valve Required by Boiler Code
Boiler Stop and Non-return Valve
Dotted Lines Indicate Piping Under Joint Jurisdiction of Boiler and Piping Codes
Second Stop Valve Required by Boiler Code
Economizer Inlet Header
Crossover Header
To Second Boiler
To Turbine
First Stop Valve Required by Boiler Code

Fig. 1—Feedwater piping which is covered by both boiler and piping codes. In this case, some 50 ft of pipe and several valves are involved

Fig. 2—Superheated steam piping in which crossover connections are involved. In case of single boiler and turbine units, all of the superheated steam piping may be interpreted to fall within the jurisdiction of the boiler code

illustrate examples of the extent of piping which is covered by the joint jurisdiction of the piping code and the boiler code.

In view of the overlapping jurisdiction of these codes, a conference of a joint committee including representatives of the ASME boiler code committee and the ASA sectional committee on code for pressure piping was held in an effort to compromise existing differences in requirements of the two codes which covered the same piping. This seemed the only feasible course to follow since the boiler code committee has been reluctant to relinquish jurisdiction over piping as defined in the boiler code.

Agreement was reached on several items, some of which were mentioned by Mr. Crocker in the January HPAC. For instance, at the present writing, values of allowable stress for use in connection with the pipe wall thickness formulas have been made the same. The boiler code committee has changed its former practice of specifying two formulas for pipe wall thickness, each applying to a certain range of pipe sizes, to agree with the piping code in which the modified Barlow formula is used throughout the complete size range. Both codes also permit the use of the alternate Lamé formula under the same conditions of size range and wall thickness. Another point of agreement concerns the permissible swing in operating pressure and temperature which is a new feature of this revision of the code and will be discussed more fully below.

The above agreement and resulting rules for both codes cover the principal points which had previously been at variance. Other items of difference are in general of less importance but involve differences of opinion or of past practice to a degree which did not permit of an immediate compromise agreement. For instance, the piping code requires that boilerfeed piping be designed for the highest pressure exerted by the boilerfeed pumps at any load under normal operation, whereas in the boiler code it must be designed for 1.25 times the lowest pressure at which any safety valve on the boiler drum is set to blow. Other points of difference are given below. It should be noted that many of these refer to items which are covered principally in the section on fabrication details of the piping code but they are applied to power piping by reference from section 1 to the fabrication detail section.

1) Special Cast Steel Fittings—The fabrication details section of the new code provides that cast specials (welding end tees, manifolds and the like) shall be equivalent in strength to the computed bursting strength of a straight section of pipe properly designed for the service. Rules for reinforcing welded branch connections also are included. The boiler code contains no design rules for determining dimensions of special cast fittings, manifolds and the like, although allowable stress values for cast materials are provided.

2) Welded Branch Connections—While the rules for welded branch connections given in the code for pressure piping were based on the rules for reinforcement of nozzle openings in the boiler code, it is understood that such rules were not intended to be used where the branch opening was more than 0.6 of the shell diameter. An investigation of the stress conditions in a full size welded branch connection and numerous bursting tests* demonstrated that the crotch section would be adequately reinforced by applying the principles given in the boiler code. The rules in the boiler code have not as yet been extended to apply to full size openings, as has been done for the code for pressure piping.

3) Welded Joints—Both codes have adopted the American Welding Society's rules for qualification of welding processes and of welding operators, but the boiler code has restrictions on the size and wall thickness of steam and boilerfeed pipe that may be welded without radiography. All welded joints in steam pipes larger than 16 in. diameter or thicker than 1⅝ in., and feedwater lines larger than 10 in. diameter or thicker than 1⅛ in., must be radiographed. No radiography is required in the piping code.

4) Marking of Welded Joints—The boiler code requires stamping all welds made by anyone other than the boiler manufacturer. The stamping must be done by an authorized inspector with a special code symbol. The piping code does not require stamping the weld with any special symbol but does require identification as follows: "After completing a welded joint, the welding operators shall stamp it with numbers or marks to identify it as their work. The weld shall be stamped with the last two numbers of the year in which it was made." Qualification of a welder in accordance with the piping code rules does not in itself permit him to stamp welds with the boiler code symbol.

5) Preheating—The boiler code does not contain any requirements regarding preheating welds of high carbon or alloy steel pipe, although many engineers hold that preheating is in some respects more important than stress relieving.

*Investigation of Stress Conditions in a Full-size Welded Branch Connection, by F. L. Everett and Arthur McCutchan, Transactions, American Society of Mechanical Engineers, FSP 60-12, Vol. 60.

The piping code, section 6, paragraph 627, reads: "Welded joints in carbon steel having a carbon content in excess of 0.35 per cent with a pipe wall thickness of ½ in. or greater, and in carbon molybdenum steel, shall be preheated adjacent to the welding zone to a temperature of not less than 400 F."

6) Stress Relieving — Paragraph P112(c) of the boiler code reads as follows: "All pipe or tube joints welded under the provision of this paragraph shall be stress relieved .....except that those in carbon steel tubes or pipe not over ⅜ in. nominal thickness, and carbon molybdenum tubes or pipe, with carbon not exceeding 0.20 per cent, not over ⅜ in. nominal thickness, need not be stress relieved."

The new piping code, section 6, paragraph 628, reads as follows: "(a) (1) Welded joints in carbon steel shall be stress relieved by one of the methods described in this section when the pipe wall thickness is ¾ in. or greater. (2) Welded joints in carbon steel having a carbon content in excess of 0.35 per cent, and in carbon molybdenum steel, shall be stress relieved when the wall thickness is ½ in. or greater. ....."

It is evident that these two sets of requirements are at considerable variance.

7) Pipe—There are a number of instances where the boiler code and code for pressure piping diverge as to requirements for pipe.

a) In Mr. Crocker's January article, it was pointed out that one of the major discrepancies between the piping code and boiler code involves the assignment of allowable stress for temperatures below 650 F. The piping code permits increased stresses on the basis that the factor of safety can be progressively lower corresponding to the reduced temperatures below 650 F. The boiler code assigns the same allowable stress for ferrous materials for all temperatures from —10 to 650 F.

b) The piping code includes a more extensive list of specifications for pipe than is covered by the boiler code. It is possible, therefore, to have a pipe line constructed of a material meeting the requirements of the code for pressure piping, but which would not be permitted by the boiler code.

c) Mr. Crocker's article discussed in detail the notes pertaining to the pipe wall thickness formula which permit using the actual measured pipe wall thickness and outside diameter in determining the allowable service pressure. The boiler code specifically requires that no thickness be used greater than the minimum as determined by application of specification tolerances.

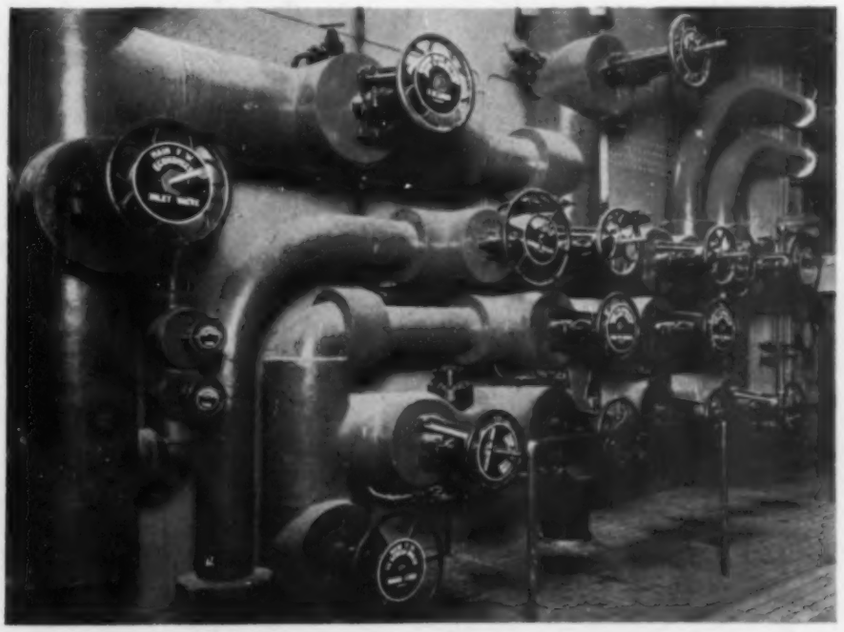
8) Design Pressures and Temperatures—The two codes are at variance as to how to arrive at the pressure and temperature intended for use in determining pipe wall thickness. The code for pressure piping uses the actual pressure and temperature existing in the line as basis for design with additional rules to govern the duration of swings above the design pressure and temperature. The boiler code, on the other hand, bases design for superheated steam piping on the superheater safety valve setting. If the operating pressure in a specific instance is very close to the upper limit of a standard valve, fitting or pipe wall thickness, then this difference of some 50 to 60 psi may become significant in high pressure plants. On feedwater piping, the boiler code ties the pipe line pressure in with the boiler drum pressure regardless of interposed economizers, static heads, etc. For blowoff piping, the boiler code requires design to conform to minimum thicknesses for feed piping whereas in the piping code, design is in accordance with the full steam pressure encountered for that portion of the pipe under full line pressure.

9) Expansion and Flexibility — Since primarily the boiler code is intended to establish rules for boiler construction, very little consideration is accorded to bending stresses in piping due to thermal expansion, dead weight between supports, requirements for hangers, supports and anchors, etc. The piping code, on the other hand, has made extensive provision for these items.

In view of these differences, the code for pressure piping will include a temporary requirement in an effort to cover that portion of the piping included under both codes as mentioned heretofore. This requirement provides that valves, fittings and piping as prescribed in the power boiler code are within the scope of the piping code except that the boiler code provisions shall govern in so far as they exceed the corresponding provisions of the piping code.

Pressure-Temperature Swings

Those familiar with operation of power plants appreciate that it is practically impossible to prevent some swings above or below the design pressure and temperature. Rapid variations in the load are responsible principally for fluctuations which, in the case of temperature, may exceed the design limits by 25 to 50 F, and the pressure may rise in the order of 50 psi above the design conditions for the superheated



Boiler feedwater valves and piping, showing identification discs on important valves, coming within the scope of the ASME boiler construction code and/or American standard pressure piping code

steam piping. Under conditions where the superheater safety valves are blowing, a pressure at least 4 per cent in excess of the design pressure at the superheater outlet may be experienced. For boilerfeed piping, the shutoff pressure on the pumps may subject the line to considerably more than 50 psi excess pressure. If the piping has been selected to exactly meet the design conditions, the stresses in the lines may exceed those permitted in the code by appreciable amounts. In fact, if the worst condition were assumed to govern design, the next heavier schedule series of pipe, and next higher pressure standard of valves and fittings, would be required in some cases.

It has been the generally accepted practice in the past, however, to design for the normal operating conditions, which has proved to be adequate from a safety standpoint. At the higher temperatures, however, strength of the low alloy materials, such as carbon molybdenum steel, decreases an appreciable amount for small increases of temperature so that it was felt desirable to place some limitation on the extent and duration such swings could attain. Accordingly, the following provisions have been included in this revision of section 7 and in the latest version of the boiler code:

a) Piping systems shall be considered safe for operation if the maximum sustained pressure and temperature on any part do not exceed the pressure and temperature allowed by this code, for all component parts of the system. It is recognized that occasional departures from the nominal operating pressure and temperature inevitably occur and therefore the piping system will also be considered safe for occasional operation for short periods at higher pressures and temperatures.

b) Either pressure or temperature, or both, may exceed the nominal design values if the computed stress in the pipe wall calculated for the pressure by formula 1 or 2. paragraph 122, does not exceed the *S* value allowable in Tables 3 and 3a for the expected temperature by more than the following allowances for the period of duration indicated:

1) Up to 15 per cent increase above the *S* value during 10 per cent of the operating period.

2) Up to 20 per cent increase above the *S* value during 1 per cent of the operating period.

Boilerfeed Systems

In the previous edition of the code, the design of the boilerfeed system was linked with the design of steam piping at a pressure somewhere within a given range of water pressures. For instance, for pressures above 1080 psi, but not in excess of 1625 psi, boilerfeed piping was to be designed as for steam at a pressure of 1500 psi. This was somewhat akin to the rules for feedwater piping specified in the boiler code as mentioned previously, wherein: (1) the feedwater piping is designed for a pressure not less than 1.25 times the lowest pressure at which any safety valve on the boiler drum or water space, is set to blow, and (2) the allowable stress used in the pipe wall thickness formula cannot exceed that permitted for the temperature of saturated steam corresponding to the set pressure of the safety valve, or the maximum water temperature to be handled, whichever is the greater.

In this revision of the piping code, straightforward rules based on the pressure and temperature existing in the line are used as the basis of design. This permits a direct evaluation of the line pressure considering such elements as static heads, economizers, etc., which are only roughly accounted for in the boiler code rules by use of the factor 1.25.

The new rules for design of boilerfeed piping are as follows:

a) The following requirements shall apply for all boilerfeed systems and systems for water, except that on boilerfeed systems the piping from the boiler drum or equivalent point to and including the feed valves required by the ASME boiler construction code shall conform to that code.

b) The designs on hot water systems shall be based on the highest pressures and temperatures actually existing in the piping under normal operation.

c) The design of boilerfeed piping systems shall be based on the highest pressures exerted by the boilerfeed pumps at any load under normal operation and on the highest corresponding temperatures actually existing. It is understood that where water passes through heaters in series, the temperature rating of the piping shall conform to the actual temperatures produced by the heaters in each part of the system.

As mentioned previously, certain swings occur, particularly on pressure in feedwater piping, which should be recognized. Design is based on highest pressures exerted at any load under normal operation, and excess pressures secured under abnormal operation are covered by the provision which permits short time swings which was quoted heretofore.

Bolting

Several new provisions for bolting have been included. Recognition of the ASTM specification for high temperature, high pressure bolting material is made. This specification covers alloy steel bolting material for temperatures from 750 to 1100 F, and includes chemical analyses and physical properties for 11 grades of material. Selection from these grades will depend on design, service conditions, physical properties and high temperature characteristics.

It is recognized that for the less severe services, a suitable grade of carbon steel bolts will be satisfactory. It is stated, therefore, that for steam service pressures not exceeding 160 psi with temperatures not exceeding 450 F, carbon steel bolts conforming to grade 22 or better of ASTM specification A107 will be acceptable. Where cast iron flanges are used, however, it is mandatory that bolting material shall be of carbon steel. It has been common experience that cast iron flanges are easily broken where alloy steel bolts are used, since the strength of the latter is out of proportion to that of the flanges. The lack of ductility of cast iron flanges is so pronounced that flanges are sometimes cracked off before any evidence of overtightening is exhibited on the part of the alloy bolts.

A further provision in this revision concerns the use of so-called "bar stock" nuts. Nuts are required to conform to ASTM specification A194 on carbon and alloy steel nuts for bolts for high pressure and high temperature service to 1100 F *ex-*

cept that nuts cut from bar stock in such a manner that their axes will be parallel to the direction of rolling of the bar *shall not be used*. Nuts of the latter type have been found unsatisfactory in piping because of their tendency to split when being pulled up to the extent required to maintain tight joints for power piping service. When washers are used, it is required that they be made of a material equal to the nut material with which they are used.

Hydrostatic Test

The requirement for hydrostatic test of welded pipe lines after erection has been revised. In the previous edition, a hydrostatic test of two times the normal service pressure was called for except that the pressure need not exceed 1½ times the primary service rating of the valves included in the piping under test. The latter requirement effectively limited the pressure to a figure below what is considered to represent a safe test for power piping with welded joints. Since the customary hydrostatic pressure test of valves is greater than twice the primary rating, the code test for welded pipe lines after erection was revised to call for twice the normal service pressure, but not to exceed twice the primary service pressure rating of valves included in the line. An exception is made for globe valves tested with the pressure under the seat, since distortion of the disc or stem due to the higher pressure could occur. The requirement as it appears in the revision is as follows:

1) Except for boilerfeed, all piping systems containing welded joints shall be capable of withstanding a hydrostatic test of at least two times the normal service pressure, to be applied where practicable. Boilerfeed piping systems shall be given a hydrostatic test of 1½ times the design pressure (paragraphs 144 and 145). The test pressure, however, shall in no case be less than 50 psi, nor more than two times the primary service pressure rating of valves included in the test. Globe valves should not be subjected to a test pressure under the seat greater than 1½ times the primary service rating, but full pressure may be applied with the pressure above the seat, or with the valve open.

Oil Piping

More rigorous provisions have been included in the new edition of the code concerning oil piping systems, as follows:

All piping systems for oil shall be designed as for steam at the same pressure and temperature. Cast iron pipe shall not be used for oil at temperatures above 300 F. Cast iron shall not be used for pipe, valves or fittings on any lubricating oil lines immediately connected to steam turbines using steam at temperatures over 500 F, or situated sufficiently near another line conveying hot material to cause danger of fire in case of failure of the oil pipe line. On all such oil lines, seamless steel pipe shall be used and all joints shall be made by welding in accordance with section 6 of this code, or joining by bolted steel flanges lapped or welded to the pipe and supplied with oilproof gaskets or ground joints. This code recognizes that in marine practice and for certain special purposes, copper tubing for lubricating oil lines is in satisfactory use but the possibility of failure due to vibration is sufficiently important to call for the above limitations except under special conditions, for which special rulings should be obtained from insurance companies or other interested parties in authority.

The extreme hazard resulting from an oil fire was considered to warrant the mandatory use of seamless steel pipe for all oil lines. Copper tubing may be used only under special conditions where individual rulings are obtained from the inspection agency in authority as noted above.

Use of Standards

Since the previous edition of the piping code, extensive revision of materials specifications and dimensional standards has been effected to keep pace with new designs and materials developed, as well as to include revisions in older methods and materials existing when the first edition of the code was issued. The sectional committee has made provision for keeping abreast of current practice in materials specifications and dimensional standards by making effective the latest revisions. The requirements governing standards and specifications in this revision of the code are as follows:

a) Unless otherwise specified, the reference "American standard," in this code shall mean the latest revision of the standard for the particular purpose as approved by the American Standards Association. The use of the latest revisions of all other standards or specifications mentioned in this code is intended.

b) The standards and specifications quoted herein are minimum requirements. Dimensions and materials, unless otherwise specified, shall be capable of meeting these standards and specifications.

c) Construction equal to or better than required by this code is mandatory, including the use of materials having physical and chemical properties at least equal to its minimum requirements.

d) Adherence to American standard dimensions for flanges and pipe fittings is strongly recommended wherever practicable. However, paragraphs, tables or notations specifying these and other dimensional standards in this code shall not be mandatory provided other designs of at least equal strength and tightness and capable of withstanding the same hydrostatic test requirements are substituted.

The revisions mentioned herein cover the more significant items of general interest, but careful perusal of the power section will disclose numerous other detail changes which have not been mentioned owing to a lack of space. The next of these articles on the code for pressure piping is planned to consider the revised section on gas and air piping.

New catalogs, bulletins, etc. are reviewed in the department "Trade Literature" which is published in HPAC monthly

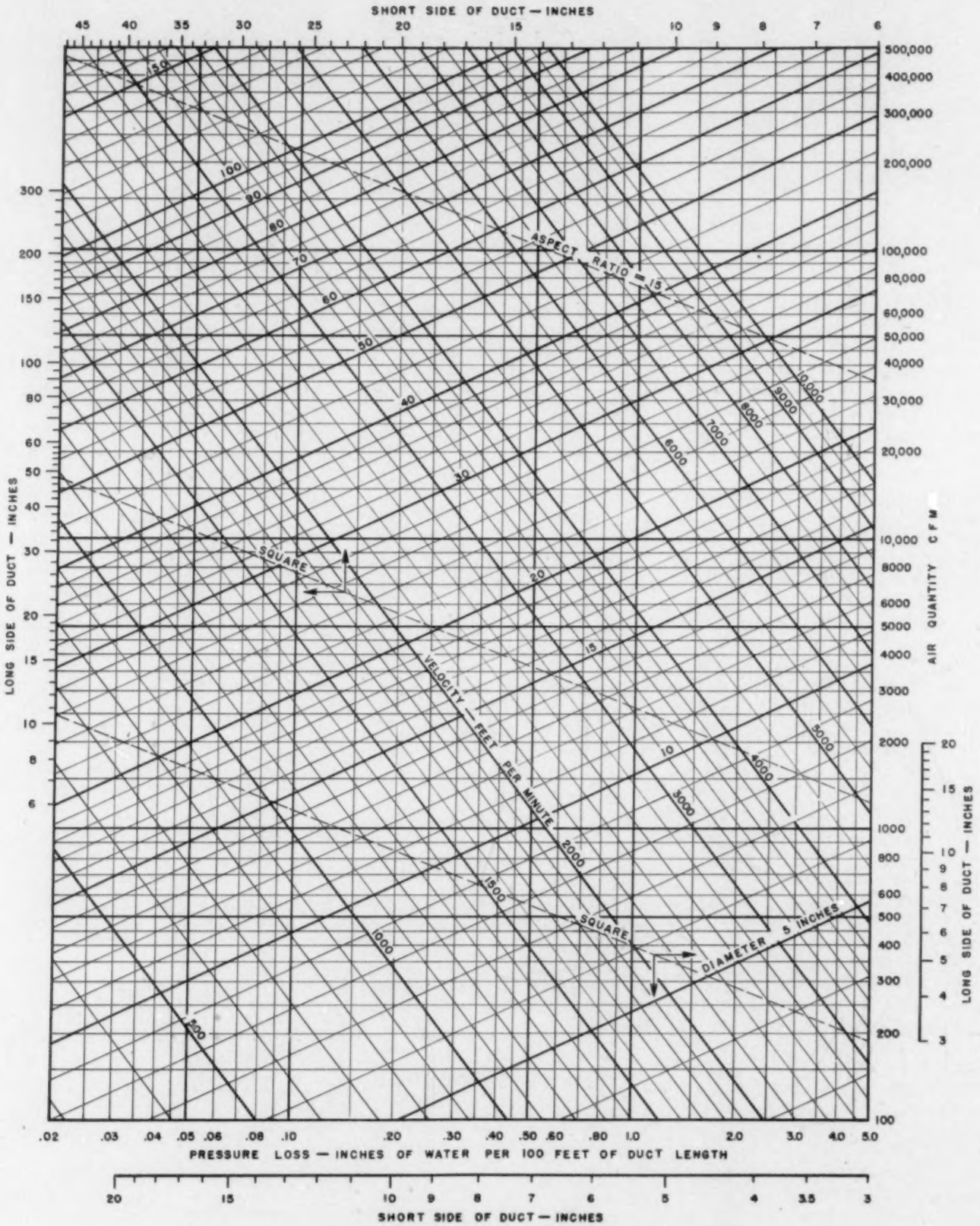


Fig. 1—Chart for determining pressure loss due to friction in ducts

How to Figure Duct Sizes

New Chart Combines Latest Data for Sizing Both Round and Rectangular Ducts—By William Goodman

THE duct sizing chart in Fig. 1 has two advantages; it offers a quick and convenient means for finding equivalent rectangular ducts; and it is based on the most recent data available.

Heretofore, auxiliary tables or charts have been necessary in order to find the dimensions of the equivalent rectangular duct. The duct sizing chart of Fig. 1 eliminates the need of these auxiliary tables and charts and permits the equivalent rectangular duct to be found quickly on the duct sizing chart itself.

Friction losses obtained from the charts commonly used today are higher than those obtained during the recent ASHVE laboratory tests. Because Fig. 1 is based on the ASHVE tests on ducts with 40 joints per 100 ft, the friction losses computed by means of it should be more nearly in accord with the losses obtained in actual installations.

The pressure loss due to friction is shown along the bottom of Fig. 1, and the air quantity along the right hand side. Lines of constant duct diameter and lines of constant air velocity are shown in the field of the chart. So far the chart is similar to other conventional charts.

In addition, the dimensions of equivalent rectangular ducts may be found on the border scales of Fig. 1. The "long side" and "short side" scales at the left and top borders respectively are used together for the larger duct sizes commonly encountered in industrial and commercial air conditioning and ventilating work. The scales at the bottom and lower right hand borders are for smaller ducts down to 3 in. in width. It should be emphasized that only the long and short side scales at the left and top are used together. Likewise only the long and short side scales at the right and bottom are used together. The two sets of scales *cannot be used interchangeably*. The range of rectangular duct dimensions shown on the chart covers those most commonly encountered. Table 1, whose use is described later, can be used for finding equivalent rectangular duct dimensions which fall outside the range shown in Fig. 1.

To find the equivalent diameter of any rectangular duct, extend an imaginary horizontal line through the long side dimension and an imaginary vertical line through the short side dimension. The diameter of the equivalent round duct is then read along the diagonal line of constant diameter through the point of intersection of the horizontal and vertical lines. Any number of different combinations of long and short sides, all of which are equivalent to each other and to the same round duct, may be read along any one given diagonal line of constant duct diameter.

The lines of constant velocity shown in Fig. 1 represent only the velocities in round ducts. The velocities in square and rectangular ducts must be calculated.

Copyright, 1942, by William Goodman.

FRICTION LOSS charts hitherto available usually have been for round ducts only. To design rectangular ducts from such friction loss charts, auxiliary tables and charts are commonly used. Anyone who has ever tried to design rectangular ducts for air conditioning and ventilating systems by means of these conventional friction loss charts for round ducts will appreciate the utility and convenience of the new chart presented here. . . . This new chart provides in one place a convenient and quick means for finding both the pressure loss and the dimensions of the equivalent rectangular duct. Designers of duct systems especially will appreciate this convenient time saving feature. In addition, in this new chart the pressure losses are based on the latest tests on ducts having 40 joints per 100 ft. Therefore, pressure losses computed by means of this chart should be more nearly in accord with the losses that are obtained in actual duct installations. . . . The author is consulting engineer, Trane Co., and also a member of HPAC's board of consulting & contributing editors

How to Use

The following examples illustrate the method of using Fig. 1.

Example 1—A duct is to carry 9000 cfm of air at a friction loss of 0.22 in. per 100 ft. Find the short side of the duct if the long side of the duct is to be 48 in.

Solution—Refer to the skeleton chart of Fig. 2. Enter the chart at point *1* which is located at 9000 cfm. Also enter the chart at point *2* which is located at 0.22 in. friction loss. Point *3* is located at the intersection of the horizontal and vertical lines through points *1* and *2* respectively. The diameter of the equivalent round duct as read on the diagonal line through point *3* is 26 in. The velocity in the equivalent round duct will be 2450 fpm.

Fig. 2—Use of chart for determining dimensions of rectangular duct by means of left and top scales

Inasmuch as the long side of the duct is to be 48 in., enter the chart at point *4* at this dimension. Extend a horizontal line from point *4* to point *5*. Point *5* is located where the horizontal line through point *4* intersects the diagonal

line representing the equivalent diameter of 26 in. Extend a vertical line from point *5* to point *6*. At point *6* read 12.8 in. as the short side of the equivalent rectangular duct.

Example 2—Take the conditions of example 1 and find the dimension of the long side if the short side is to be 20 in.

Solution—Locate 20 in. on the short side scale at the top of the chart. Find the intersection of an imaginary vertical line through 20 in. and the diagonal line of 26 in. diameter. Through this point of intersection, extend a horizontal line to the long side scale at the left of the chart. Read 28.5 in. as the required dimension of the long side.

Note in both of the preceding examples that the same equivalent diameter of 26 in. is used. The equivalent diameter is the same as long as the air quantity and pressure drop are the same. However, there are any number of equivalent rectangular ducts corresponding to the one diameter of 26 in. For the preceding two examples, the dimensions of all of these various equivalent rectangular ducts would be read along the one diameter line of 26 in.

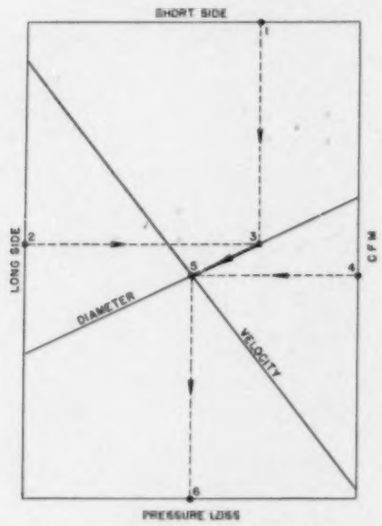


Fig. 3—Use of chart for determining pressure loss due to friction

Example 3—Six thousand cfm of air is flowing through a 40 in. by 16 in. rectangular duct which is 250 ft long. Find the pressure drop in this duct.

Solution—Referring to the skeleton chart of Fig. 3, locate point *1* at 16 in. and locate point *2* at 40 in. Extend vertical and horizontal lines through these two points respectively. These two lines intersect at point *3*. The diameter of the equivalent round duct as read at point *3* is 27 in.

Enter the chart at point *4* at 6000 cfm. Extend a horizontal line through point *4* until it intersects the diagonal line for the 27 in. round duct at point *5*. Drop a vertical line from point *5* to point *6*. The friction loss in the 40 in. by 16 in. rectangular duct is 0.088 in. per 100 ft as read at point *6*. The friction loss in the entire length of duct is (250/100) × 0.088, or 0.22 in. of water.

The scales of rectangular duct dimensions at the bottom and right side of Fig. 1 are used in exactly the same manner as those at the left and top. After first locating the diagonal line of diameter, any number of different combinations of short and long sides can be read on

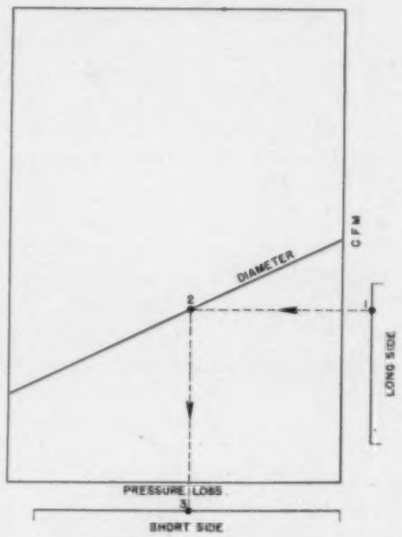


Fig. 4—Use of chart for determining dimensions of rectangular duct by means of bottom and right scales

the bottom and right hand scales. In all cases, the imaginary vertical and horizontal lines through the short and long sides must, of course, intersect on the one diagonal line representing the given diameter.

The method of using the bottom and right hand scales is illustrated in Fig. 4. The diameter of the equivalent round duct is represented by the diagonal line shown. Locate point *1* at the desired dimension for the long side of the equivalent rectangular duct. Project a horizontal line from point *1* until it intersects the diagonal line of given diameter at point *2*. Drop a vertical from point *2* to the short side scale. At point *3* read the required dimension of the short side of the duct. The process can, of course, be reversed and the long side found for a given short side and diameter.

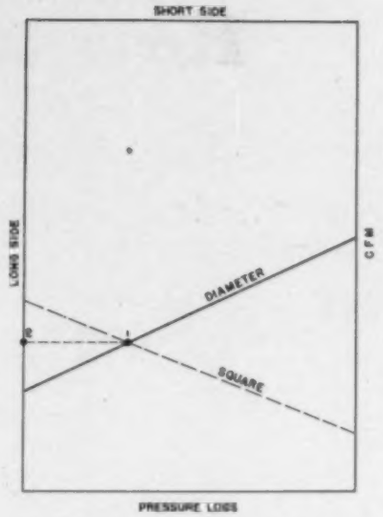


Fig. 5—Use of chart for determining dimensions of square duct

Square Ducts

The two dashed diagonal lines marked "square" in the lower portion of Fig. 1 are used for finding the size of *square* duct equivalent to any given diameter of round duct. As the arrows on these lines indicate, the upper square line is used with the left and top scales of rectangular duct dimensions, and the lower square line is used with the right and bottom scales.

The dimension of the equivalent square duct for any given diameter is found at the intersection of the square line with the line representing the given diameter. Thus in Fig. 5 point *1* represents the intersection of the solid line of 20 in. duct diameter with the square line. Extend a horizontal line through point *1* to the left scale, and read there 18 in. as the dimension of the square duct equivalent to the 20 in. round duct. The size of the square duct can also be found by extending from point *1* a vertical line up to the short side scale at the top of the chart. Use whichever scale is most convenient; the answer will, of course, be the same with either scale.

The lower square line in Fig. 1 is used in exactly the same way as the upper one except that it is used with either the bottom or the right hand scales of rectangular duct dimensions.

The square duct equivalent to a given round duct can also be found with a slide rule. The diameter of a round duct is 10 per cent greater.

than the dimension of the equivalent square duct. Hence, if the diameter is given, the equivalent square duct can be found by dividing the given diameter by 1.1. Thus for the 20 in. round duct mentioned in a preceding paragraph, (20/1.1) = 18 in., the dimension of the equivalent square duct.

Aspect Ratios

A dashed line marked "aspect ratio = 15" slopes downward across the top of Fig. 1. Any combination of rectangular duct dimensions which intersect on this line will have a ratio of 15 to one, that is, the long side of the duct will be 15 times the short side. For any point of intersection below this dashed line, the aspect ratio will be less than 15; for any point above, greater than 15. The long side and short side scales should not be used for any duct whose aspect ratio is greater than approximately 15. Fortunately this is not a serious limitation because it is very rare that ducts having an aspect ratio as extreme as 15 are used in actual practice.

The values in Table 1 may be used for ducts of any aspect ratio.

Example 4—A 24 in. diameter duct is required. If the long side of the equivalent rectangular duct may be 48 in., find the dimension of the short side.

Solution—

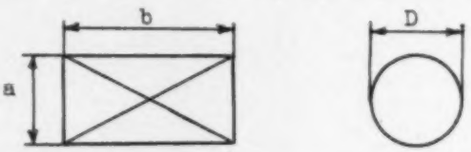
$$\frac{D}{b} = \frac{24}{48} = 0.50$$

In Table 1, opposite the value 0.50 in the column headed "*D/b*" find $a/b = 0.23$. Hence, $a = 0.23 \times 48 = 11$ in.

Symbols

a = short side of rectangular duct, in.;
b = long side of rectangular duct, in.;
d = density of air, lb per cu ft;
D = diameter of round duct, in.;
N = fan speed, rpm;
p = pressure loss due to friction, in. of water per 100 ft of duct length.

Table 1—Ratios for Computing Equivalent Round Ducts



a/b	D/b	a/b	D/b	a/b	D/b
0.01	0.079	0.36	0.644	0.71	0.925
.02	.120	.37	.654	.72	.932
.03	.153	.38	.664	.73	.938
.04	.182	.39	.673	.74	.946
.05	.207	.40	.683	.75	.951
0.06	0.231	0.41	0.691	0.76	0.958
.07	.252	.42	.700	.77	.965
.08	.273	.43	.709	.78	.970
.09	.293	.44	.719	.79	.977
.10	.312	.45	.727	.80	.983
0.11	0.329	0.46	0.736	0.81	0.990
.12	.346	.47	.744	.82	0.995
.13	.362	.48	.753	.83	1.002
.14	.379	.49	.761	.84	1.009
.15	.393	.50	.770	.85	1.014
0.16	0.409	0.51	0.777	0.86	1.020
.17	.423	.52	.786	.87	1.026
.18	.437	.53	.793	.88	1.032
.19	.450	.54	.802	.89	1.038
.20	.464	.55	.809	.90	1.045
0.21	0.477	0.56	0.817	0.91	1.049
.22	.490	.57	.825	.92	1.055
.23	.502	.58	.832	.93	1.061
.24	.514	.59	.840	.94	1.067
.25	.526	.60	.847	.95	1.074
0.26	0.537	0.61	0.854	0.96	1.079
.27	.549	.62	.862	.97	1.083
.28	.560	.63	.869	.98	1.090
.29	.571	.64	.876	.99	1.096
.30	.582	.65	.884	1.00	1.101
0.31	0.593	0.66	0.890		
.32	.603	.67	.897		
.33	.614	.68	.905		
.34	.624	.69	.912		
.35	.634	.70	.918		

Air Density

Fig. 1 is based on standard air whose density is 0.075 lb per cu ft. Usually the density of the air flowing through a duct system will differ from that of standard air. In this case, the friction loss through the duct as actually measured by a manometer will be different from the friction loss that would be obtained if standard air were flowing through the duct.

The effect of air density on the pressure loss in a duct system must be considered from two separate viewpoints: (1) the design of the duct system; and (2) testing of the duct system.

Case 1—All performance curves for fans and other equipment used in air distribution systems are universally rated for standard air whose density is 0.075 lb per cu ft. Consequently, when computing the friction loss in duct systems, such friction losses should be based on the density of standard air regardless of the actual density of the air.

If the actual friction loss through the duct system were to be computed on the basis of the actual density of the air flowing through the duct system, this actual friction loss would afterward have to be corrected to the friction loss that would take place if standard air were flowing through the duct system. This

being the case, it is useless to compute the friction loss by using the actual density of the air. Instead, the friction loss should be computed on the basis of standard air by means of Fig. 1. After the duct friction loss with standard air has been determined by means of Fig. 1, the fan can be selected from the rating tables on the basis of the static pressures with standard air and the volume of standard air required. After the fan has thus been selected, its speed can be corrected by means of the following formula:

$$\frac{N_2}{N_1} = \frac{d_1}{d_2} \quad \text{.....[1]}$$

If the duct friction is estimated from Fig. 1, the fan is selected in the manner described in the preceding paragraph, and the speed is corrected by means of equation 1, the required weight of air will be delivered regardless of the actual density of the air.

Notice in the preceding paragraph that the emphasis has been placed on delivering a given weight of air through the duct system rather than a given volume. When designing a duct system for air conditioning or heating, the weight of air to be delivered to the room—not the volume—is of prime importance. Regardless of the density of the air, the required weight of air must be delivered to the room if proper temperatures and humidities are to be maintained.

Case 2—During tests on ventilating and air conditioning systems, the duct friction loss must frequently be measured. When measuring the friction loss in a duct system through which nonstandard air is flowing, the pressure loss measured by the manometer will differ from the pressure loss computed by means of Fig. 1. If the friction loss has been computed on the basis of standard air, the pressure loss that will be obtained when the *same volume* of nonstandard air is flowing through the duct can be computed by means of the following equation.

$$\frac{p_2}{p_1} = \frac{d_2}{d_1} \quad \text{.....[2]}$$

Conversely, equation 2 can also be used to convert the measured friction loss back to the pressure loss which would have been obtained had the *same volume* of standard air been flowing through the duct.

"OPEN FOR DISCUSSION"

Communication to the Editor

PANEL HEATING'S VALUE PROVED BY USE

THE EDITOR—

It was with considerable interest that I read J. H. Walker's comments on panel heating on pp. 751-752 of the December HPAC. It seems to me, however, that there still exists a certain amount of confusion on this subject because many people continue to look upon radiant heating as simply another method of heating a building, instead of studying its effect on the human mind and body. I notice that Mr. Walker suggests that with radiant heating the overall feeling of warmth will be unchanged, but, in my opinion, we do not need a feeling of warmth. When the conditions are such that we have no feeling of either heat or cold, the state of the body will be more tranquil and the mind will not be conscious of any kind of strain on our thermostatic system. If the body loses more of the dissipated heat by radiation and less by convection than our thermostatic system is ordained for, then the mind is undoubtedly conscious of a strain, whether the individual knows the reason or not.

Everyone has undoubtedly experienced at various times the mental stress which takes place when finding himself trying to work in an atmosphere where the air temperature is too high; and again, most people have experienced that extra agility and alertness which develops when one is working or walking in a nice sunny place where the air temperature can be considered on the chilly side. I think it is sufficiently well established for me to say with confidence that it is the radiant energy direct from the sun, plus the cooler and longer wave length rays reflected from surrounding objects, which protect the body from too much heat loss; and that it is the cooler air which creates alertness and promotes the extra energy from within.

It may be that we shall have to wait a few more years before we learn in what way the cooler air promotes greater energy and if it actually promotes a greater generation of heat within, but it does seem true that when we are in a cooler atmosphere, our appetites develop, indicating that something within is being used up more rapidly than when we are in an atmosphere of higher temperature. One might compare this with a fire which needs very little fuel when it is burning sluggishly. However, as soon as more draft is applied and the fire burns more rapidly, then fuel must be added at a greater rate.

It may be correct that some laboratory investigations did not throw any light on this very interesting point, but I do not think it is wise to say—without further investigation—that there is no truth in the theory. I realize the very great value of laboratory investigations, but I am one of those who feel we should be very careful in accepting as final anything concerning the reaction of a condition on the human body until it has been verified in the actual conditions under which it has to operate.

I have spent many years experimenting with different methods of heating, both in the laboratory and in the field, and have come to the conclusion that many laboratory tests should be taken as a guide only until the results have been substantiated in the home, office, school and other buildings. On the whole, the results obtained under normal conditions are more reliable than specially prepared laboratory tests. Laboratory tests are invaluable, and one may say essential, for obtaining some data; but when searching for information concerning the general

[*Concluded on p.* 113]

Steel Production Aided by
AIR CONDITIONED CRANE CABS

One Crane Operator Now Does the Work of Three ..... By Henry W. Moore

CABS of cranes working directly over steel mill pits which are at a temperature of 2400 F are hot spots. Average air temperature in the cabs may be 165 F; thus three crane operators working 20 min each hour may be required per shift. . . . At the steel mill described here, the crane cabs have been air conditioned, and now one operator can work continuously during an entire 8 hr shift. Saving in labor cost was not the principal reason for the installation, however; there is a shortage of skilled crane operators; and also 20 min of 165 F temperature an hour is an uncomfortable and probably unhealthy condition. . . . In order to cool the cabs, they had to be enclosed. This brought up the problem of communication between the operator and ground crew, which was solved by means of two-way radio. An air cooled condensing unit is installed on the top of each cab, serving an air conditioning unit in the cab. Operation of air cooled condensers and motors in the high temperatures encountered brought up some special problems, the solutions of which are described here by Mr. Moore, manager of the air conditioning engineering department of the Bimel Co. Some trouble has also been had with the thermostatic control, due to jarring when the cabs start and stop suddenly

AN unusual application of air conditioning was recently completed at one of the steel companies. This installation is for cooling two operators' cabs on cranes which handle steel ingots directly over the pits, which are at a temperature of approximately 2400 F.

Until these new air conditioned crane cabs were installed, a workman could work no longer than 20 min in the heat which, at times, reaches nearly 200 F. The average temperature of the air above the pits, however, is about 165 F. Since the new, totally enclosed air conditioned cabs have been installed, one operator is now able to work the full 8 hr shift. In fact, it has been reported that some of the operators now eat their lunches in the cabs as they are more comfortable there than anywhere else around the plant. Formerly, three operators were required on each shift, each man working 20 min of each hour.

While cooled crane cabs give the company a tremendous saving in labor costs—enough to pay the cost in less than a year—the installation was not made for that purpose alone. There is a shortage of skilled crane operators and as each operator is three times as productive as formerly, the war production program is aided. Also, as mentioned, the terrible discomfort and unhealthy condition encountered by one working in such a high temperature as 165 have been done away with.

Cab Temperatures

At the time the cooling system was designed, no similar job had been installed where the ambient temperature averaged over 125F—to my knowledge at least. However, there had been a few installations prior to this one, but at lower temperatures. We were told by the owners that they would be satisfied if the cab temperature were not lower than 125, which would be a 40 F temperature reduction below the surrounding temperature of 165 F. However, we designed to produce an inside condition of 105, that is, a 60 F temperature difference. In actual operation, we find that we are now able to maintain the temperature in the cabs at 95 F or lower, which seems to be quite necessary in order to keep the operators comfortable. Most of the time the cab temperature averages about 85, which of course is a wonderful condition compared to the 165 F temperature formerly encountered.

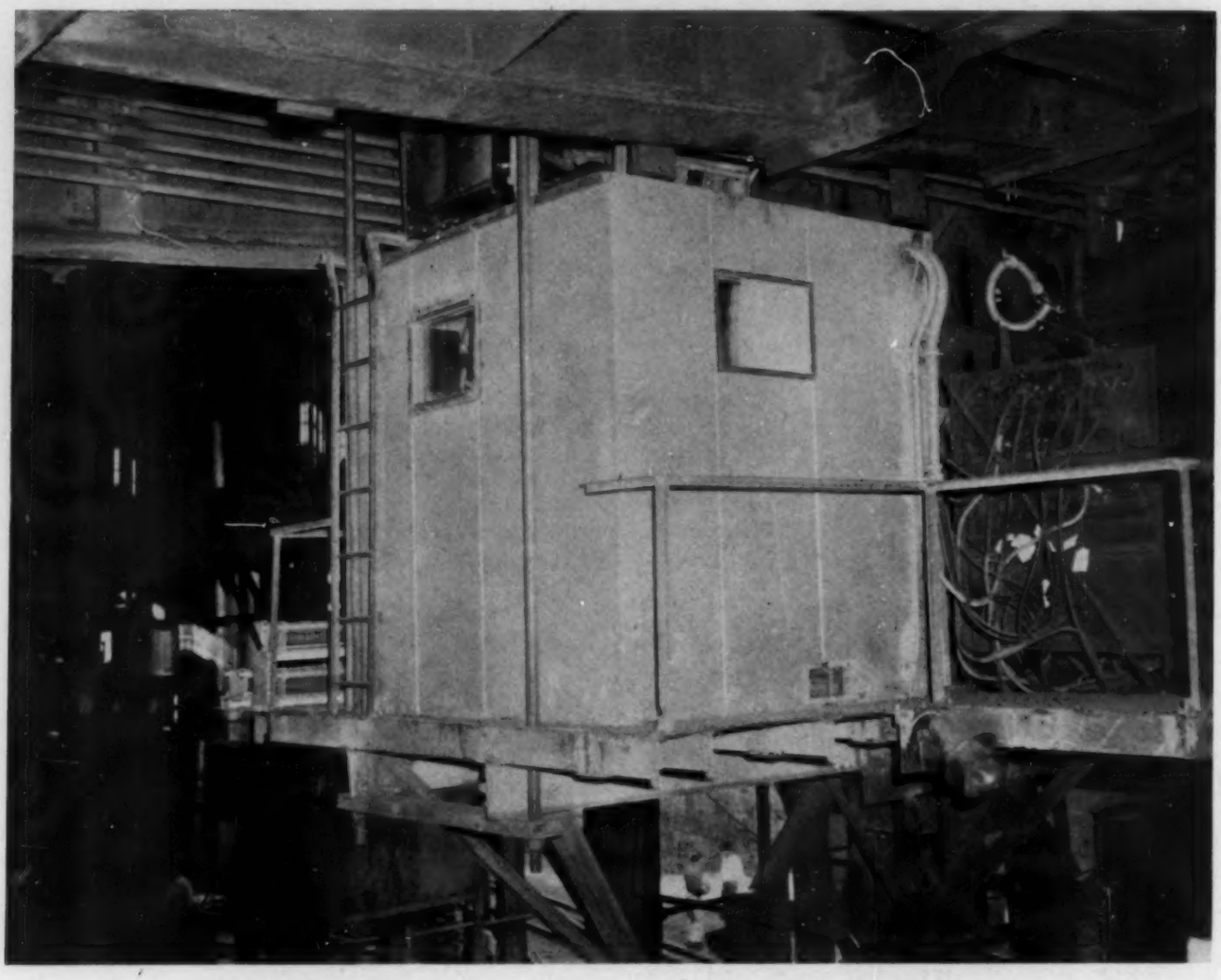
Cabs Enclosed

Before the cabs could be conditioned it was of course necessary that they be totally enclosed; formerly, they were open so that the operators could get as much ventilation as possible. As it was not practical to enclose the old cabs, new ones were built. The new ones are each 7 ft long, 7 ft wide and 7 feet high, all surfaces being thoroughly insulated. Floors are insulated with 4 in. rock cork and the other surfaces with 2 in. of the same material. The outsides of the cabs were painted aluminum to reduce the radiant heat from the steel pits. Windows were provided, 22 in. by 16 in., and there is a glass covered observation hole for the floor of the cab approximately the same size as the windows.

Enclosure of the cabs brought up another problem—satisfactory communication from operator to ground crew. Hand signals had been used, but it was found necessary to install two-way radio communication. This was done by the steel mill, and solved the communication problem very satisfactorily.

Air Cooled Units

One stipulation which the owners insisted upon—and which, at first, made the job seem almost impossible—was that no water be used in connection with the cooling equipment. Of course, it was quite impractical to use water for condensing purposes for the refrigerating ma-



He floats through the air with the greatest of ease, the operator in this air conditioned crane cab at a steel mill. High cab temperatures formerly made it necessary to have three operators per shift, each working 20 min out of each hour. Now, one operator can work the full 8 hr shift, usually chooses to have lunch in his cab as well, as it's the most comfortable spot around. Communication with the ground crew is by radio

chine, as the crane moves about the plant in an area approximately 600 ft long by 100 ft wide. This meant that an air cooled condensing unit had to be used, and it can readily be appreciated what a problem this presented, using 165 F air for the condensing medium.

The manufacturer who cooperated with the writer had no exact data on the operation of his condensing units at such a high condensing temperature. In actual operation, it was found that the condensing pressure averages about 250 psi gage.

The uncertainty of the head pressure made it difficult to determine the size of the compressor motor required. A 2 hp motor was decided upon and used, but at the start-up of the system, it was found that this motor was slightly overloaded and hence we added an auxiliary condenser (together with a propeller fan for ventilating it) which lowered the condensing pressure to about the figure mentioned above and which made the 2 hp motor ample in capacity.

Special Parts

At this point, it might be stated that practically every item in this system was specially built, as one might readily imagine, due to the high temperature. A special built-up air conditioning unit, approximately 54 in. long by 18 in. wide by 54 in. high, was placed inside each cab alongside one wall, with the small air intake through the wall. The unit is equipped with an eight row, direct expansion cooling coil, with a special high pressure thermostatic expansion valve. The fan used is a direct connected 700 cfm blower driven by a special ¼ hp, 860 rpm, high temperature motor, the available current being 230 volts d-c. The air conditioner has two permanent washable air filter cells. The supply air is distributed through a 16 in. by 12 in. adjustable directional register, situated on top of the air conditioning unit.

The 2 hp "Freon" refrigerating compressor, with the air cooled condenser, is on top of the cab, as shown in the photograph. (This picture was taken before the auxiliary condenser was added.) Of course, the condensing unit motor is a special high temperature design, similar to the one for the fan mentioned above. This condensing unit was

made special for this job, being equipped with many special parts, such as bearings, gaskets, belts, fusible plugs, oil and pressure switch. There are vibration elimination joints in the copper refrigerant lines connecting the condensing unit with the cooling coil.

Some "Headaches"

The results from the operation of the system are very definitely known as this job has been in use since April last year and ran through the entire summer of 1941 with recording thermometers accurately measuring the temperatures in each cab constantly. The steel mill states that unquestionably the job has been a success, the cooling results being even better than anticipated.

However, there have been a few "headaches" in connection with this system, as would be expected with any more or less experimental type installation such as this. The first difficulty we encountered, as mentioned above, was the necessity for doubling the amount of condensing surface and adding an auxiliary ventilating fan for that extra condenser surface. This change greatly improved the performance of the system. Incidentally, this is the only important change which has been made in the equipment since installation. The biggest trouble, to date, has been with the special high temperature 2 hp direct current motors on the condensing units. However, new motors of a different design have been furnished recently and the manufacturer expects these to relieve the trouble.

Another difficulty has been with the thermostat installed in each cab, which tends to make and break the electrical starter circuit unnecessarily. At first, one might wonder why it would be necessary to have a thermostat in such a system. However, it has been found desirable because there are times when the cab is not over the crane pit, particularly during certain hours of the night shift, when the refrigerating capacity is much greater than required. It then lowers the temperature in the cab to as low as 55 F at times, which is too cold for the operator. Hence it is desirable to have a thermostat as a low limit control, which is usually set at 80 F.

So far, no perfect answer has been found to this thermostat requirement, because it should be one that is absolutely "jarproof," due to the extreme shaking of the cab as it starts and stops suddenly, and also an instrument which will stand temperatures as high as 250 F, which may be encountered should the system fail temporarily while over the steel pit. Furthermore, this instrument must have a reasonably close temperature differential. No thermostat has been found which is entirely satisfactory for this application. However, we are using one suspended on a rubber swivel which permits it to move in all directions and seems to be giving fair results, although at times it still seems to break the circuit, due to jarring. Of course, if necessary, the operation of the compressor can be made manual instead of automatic.

This application of air conditioning certainly seems to be another distinct contribution by the air conditioning industry to the war program for steel production, and one which probably should be used throughout the hot steel mills of the country. It would seem that such an installation would be very profitable and advisable on all cranes which handle steel ingots over open pits.

[*Concluded from p.* 110]

reaction of any kind of heating on the human mind and body, we have to collect a lot of data spread over several years, and it appears to me that there is nothing like searching for this in the places where most of the ordinary people are found. During the years of my research work I invariably used the laboratory to obtain information which acted as a guide, but before actually accepting such findings, I had several systems installed and tests made on the job so that the results could be checked to verify or modify laboratory findings. Before we decided on the special floor construction for the Liverpool cathedral, which I believe is the largest heated floor in the world, I spent over a year experimenting with different types of flooring and the results of this kind of radiant heat on a large number of persons. It has been my experience that when people are selected for a laboratory test, they invariably let their minds dwell on the things which they think should happen, instead of relaxing and allowing the conditions to react on them as they would do in a normal way.

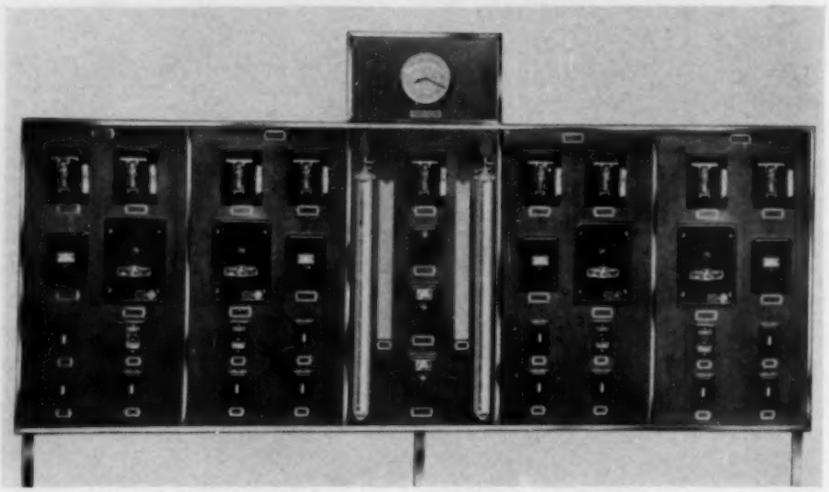
I am not surprised that some laboratory tests appear to give negative results, but I am surprised that negative results without substantiation are sometimes accepted as a basis for a theory which often proves to be an erroneous one, the result being that it is only after years of experience that the truth is eventually revealed. Isaac Watt, the great inventor, experimented with radiant heat over 150 years ago and gave up the idea because his results were negative. We now know that the only reason his results were negative was that he used polished surfaces instead of a dull rough surface; but we had to wait over 120 years for radiant heat to be introduced because a negative result was accepted as final. I am afraid that it is because many people are apt to accept laboratory tests as final that we still do not have a perfectly reliable system for health and comfort; and it is partly that reason why we get so much erroneous data and factors placed before us. I am sometimes astonished when reading articles which disclose how, from results of a few meager tests and isolated cases, assumptions of great moment are made and deductions formed.

While radiant heat may appear to be something new in this country and create some doubt in the minds of some engineers, I think it should be remembered that heating engineers in Europe have had over 30 years of experience with this kind of heating. While from this experience we know something of its advantages and reactions, we certainly would not suggest that radiant heat panels alone are the answers to all kinds of heating and air conditioning problems. I do, however, believe that more healthful and efficient systems are produced when, in designing a system, we pay more attention to the way heat escapes from the human body than simply to provide an apparatus which will raise the temperature of the air in a building to a predetermined degree—T. Napier Adlam, chief engineer, Sarco Co., Inc.

Design of an Oil Fired HIGH PRESSURE STEAM PLANT

By Kalman Steiner

BUNKER C (or No. 6) fuel oil is generally considered to average 152,000 Btu per gal. At 77 per cent efficiency there will be available in useful heat absorbed by the boiler a total of 117,000 Btu per gal of oil burned. From the requirement of 1055.85 Btu per lb water evaporated, stated in my January article, it will be necessary to burn very close to 406 gal per hr to provide for a maximum steam demand of 45,000 lb per hr. Parenthetically, it might be remarked here that this fuel ratio would establish a fuel cost for steam as about 31.5c per 1000 lb, assuming fuel oil to cost 3.5c per gal. However, it is more customary to express cost of steam on the basis of "commercial" steam, meaning on a basis of "from and at" 212 F. Substituting a heat content in the steam of 970.4 Btu instead of 1055.85, the fuel cost for commercial steam would be 29c per 1000 lb.



Typical panelboard for a four burner installation

Burner Types

Some four distinct types of burners can be applied to the boilers. The selection of one of them must be based on the type of operation desired. To minimize the labor and attention on the part of the engineer, leaving him free to work on other duties about the boiler room, it is possible to install full automatic burners; these would be motor driven rotary burners. If the rotary burner principle is desired without the fully automatic feature, the burners can be semi-automatic with manual ignition, and in that case can be either motor driven or steam turbine driven. To dispense with rotating mechanisms, and retain semi-automatic control, the burners could be mechanical pressure atomizing. The fourth possibility would be steam atomizing, but this would probably be the least likely choice, because of the amount of steam used through the nozzle.

Pumps and Heaters

Should fully automatic burners be selected, then it would be well to make the pumps motor driven as well as the burners. In the event of semi-automatic burners, then the pumps could be either motor driven or steam driven, although it is imperative to provide at least one motor driven pump for starting up cold boilers. For the same reason, should steam turbine burners or steam atomizing burners be chosen, at least one motor driven burner will be needed for starting up in the absence of steam.

The oil heaters can be steam operated, regardless of which of the four burner types is selected, and an auxiliary electric heater should go in the system, large enough to provide sufficient oil at suitable temperature to operate at least one burner at maximum capacity should steam be lacking. The heating coils of the storage tanks can utilize either steam or hot water as the heating medium. Steam is the more conventional, but hot water obtained from a small heat exchanger makes an ideal heating medium as its circulation can be controlled very accurately from a pump and thermostat so as to maintain proper oil temperature in the tanks. For very long runs of suction line it would be well to consider an electric heating system for the oil pipes themselves,

STEAM PLANT — Mr. Steiner (consulting engineer for the Ray Heat & Power Co., and a member of HPAC's board of consulting & contributing editors) considered in the January HPAC the choice of fuel, determining the load and the load factor, boiler capacities, boiler types and boiler selection. This month, he covers oil burning, piping, pumping and storage. . . . His articles are based in part on the design used for a number of recent national defense construction projects outside the continental United States

which is a means of heating up the metal of the pipe itself by passing a low voltage current through it.

Experience has shown the desirability of pumping at least 50 per cent more oil than will be burned at maximum load. This sets up a circulating oil stream from which the burners can withdraw the amount to be burned at any time for the prevailing load condition. Surplus oil over burner requirements can be either returned to storage or recirculated back into the pump suction. Usual practice is to install a valve system that will permit diverting the return oil at will, the oil usually being returned to storage. Should some operating condition necessitate quick raising of oil temperature, then the return oil can be recirculated.

Pumps and heaters are generally installed in duplicate, each sized for full load operation. In the conventional circuit, oil is drawn from storage by pump suction, strained, discharged through the heater and then conducted to the burners. As mentioned, surplus oil over burner firing rate is conducted back to storage or recirculated, as desired.

Number of Burners

For the case of a single water tube boiler carrying the entire load, the number of burners will depend somewhat upon the burner type and the draft. If mechanical pressure atomizers are used, it will be possible to burn about 125 gph per burner on natural draft, and so four burners should be installed. Under forced draft, a capacity of 200 is feasible, and so three burners will suffice. As to rotaries, these would be used on either natural or induced draft—or more rarely, and in special cases only, forced draft. Three burners would probably make the best arrangement on this boiler, although it could be done with two, with either natural or mechanical draft.

Fire tube boilers of the size indicated can easily be fired with a burner per boiler, regardless of burner type or draft. A boiler containing 1800 sq ft of heating surface had best be equipped with a 100 gph burner; there being six such burners the total burning capacity will be 600 gph. Or if five 2500 sq ft boilers go in, each of these will require a burner rated at 140 gph, making a total burner capacity of 700 gph.

It now appears that whereas we started out to design a system capable of burning 406 gph, we now have laid out one considerably larger. This comes about principally from two causes: in selecting boilers it is necessary to utilize commercial sizes coming reasonably close to design sizes, and in assigning a burner capacity it is only logical to allow ample surplus. In fact, as the steam generating rate per sq ft of boiler heating surface is increased, the margin by which burner ultimate capacity exceeds probable maximum required capacity must be increased. That is to say, if a boiler is operated to evaporate at a rate of 3.5 lb per sq ft per hr, a burner capacity of 0.3 gal per hr for each sq ft will be ample. When the evaporating rate is doubled, the burner capacity will have to be increased to about 0.75 gal. There is no definite ratio of rate of increase between the two; rather it is a matter of judgment.

Pipe Sizes

Sizes of pipe for oil suction, oil discharge and oil circulation at various points in the system, as well as size of steam and condensate connections to the oil heaters, can all be calculated from fundamental data. This would be somewhat tedious, and in most cases unnecessary. Equipment manufacturers usually issue dependable data of this nature. A summary of pipe sizes is given in the table below. Strainers should be of the same pipe size as the line in which they are inserted. Valves can usually be of the smaller pipe size if they are situated at a point where reduction of pipe size occurs. For instance, if a 2½ in. suction line is used with a 1¼ in. pump, the strainer should be 2½ in. but the valves may be 1¼ in. It is also desirable to have a valve on the tank side of the strainer, which of course would have to be full suction line size.

Intelligent operation of a plant is most difficult if there are not a sufficient number of instruments in the oil flow system. There should be

Nominal Pipe Sizes for Oil and Steam, Inches

(Oil lines based on pumping No. 6 commercial standard, at 100 F in suction lines and 160 F in circulating and return lines. Steam lines based on initial pressure of 100 psi)

Gph	Pipe	Length of Pipe Run, Feet							
		50	100	150	200	250	300	400	500
100	S	1½	1½	1½	2	2	2	.....	.....
	D	1¼	1½	1½	1½	2	2	2	2
	St	½	¾	1	1¼	.....	.....	.....	.....
200	S	2	2	2	2	2	2	.....	.....
	D	1½	1½	1½	2	2	2	2½	2½
	St	½	¾	1	1¼	.....	.....	.....	.....
300	S	2	2	2	2	2½	2½	.....	.....
	D	2	2	2	2	2	2	2½	2½
	St	¾	1	1¼	1½	.....	.....	.....	.....
400	S	2	2	2	2½	2½	2½	.....	.....
	D	2	2	2	2	2	2	2½	2½
	St	¾	1	1¼	1½	.....	.....	.....	.....
500	S	2	2	2½	2½	2½	3	.....	.....
	D	2	2	2	2	2	2½	2½	3
	St	¾	1	1¼	1½	.....	.....	.....	.....
750	S	2	2½	2½	2½	2½	3	.....	.....
	D	2	2	2½	2½	2½	2½	3	3
	St	1	1¼	1½	2	.....	.....	.....	.....
1000	S	2½	2½	2½	2½	3	3	.....	.....
	D	2	2	2½	2½	2½	2½	3	3
	St	1	1¼	1½	2	.....	.....	.....	.....
1500	S	2½	2½	3	3	3	4	.....	.....
	D	2½	2½	2½	2½	3	3	3	3
	St	1	1¼	1½	2	.....	.....	.....	.....
2000	S	3	3	3	3	4	4	.....	.....
	D	2½	2½	3	3	3	3	3	4
	St	1¼	1½	2	2½	.....	.....	.....	.....
2500	S	3	3	3	3	4	4	.....	.....
	D	3	3	3	3	3	3	4	4
	St	1¼	1½	2	2½	.....	.....	.....	.....

S = Suction line, between pump and storage tank.
D = Discharge line, including circulating line from pump, through heaters, to burners and back to suction or return to tank. Each portion, including the return, is separately sized from the table.
St = Steam line, from header to preheater.
Suction lines are not recommended to exceed 300 ft.
Length of pipe refers only to run of pipe in one size; when a branch occurs begin a new run.
Thermostatic steam valve may be same size as pipe indicated for run of 50 ft steam.

a vacuum gage on each side of the suction strainer, and a pressure gage on each pump discharge and at each burner. A thermometer will be needed in the tank suction, at the heater outlet and at each burner.

Automatic Ignition

It has already been pointed out that by utilizing automatic ignition the amount of attention required of the engineering staff is greatly reduced. Even though a plant be isolated from a community so that fuel gas from a utility supply is not available for ignition, it is quite a simple matter to install a system of bottled gas for that purpose. There should be two tanks, with suitable pressure regulating devices and an automatic cutover to cut in a fresh tank when one becomes empty. A sensitive pressure control will be needed in the gas supply line either to operate an alarm or cut off the electric supply to the burners when the gas pressure becomes too low. Since automatic burners are generally provided with electric-gas ignition, gas is actually burned only for a minute or two each time the burner starts. In addition, such burners are usually arranged also for graduated flame control, whereby the firing rate is proportioned to the load, so the number of stops and starts is low, assuring that gas tanks will not need frequent replenishing.

Whether ignition be automatic or manual, there should be complete safety protection against flame failure. The most complete protection is afforded by photelectric devices. By means of these, oil supply to the burner can be snapped off instantly should the flame die out. Moreover, on automatic ignition, the safety arrangement can be even more complete, in that the ignition gas flame can be supervised by electronic means as well as the main oil flame. Practically, this means that the oil supply is not turned on until the gas pilot is properly burning.

Draft Control

Regardless of the control style, all burners will perform at maximum efficiency only when the draft is rigidly controlled. There are a number of methods for accomplishing this: barometric, pneumatic and electric. The barometic is the simplest and least costly, but its use is generally confined to smaller installations of but one or two boilers.

The pneumatic is most complete, and is available in a wide variety of forms. For multiple boiler installations, a very satisfactory system would comprise a master controller, the motion of which would be governed by changes in load, and which would in turn send proper impulses to fuel feed controls, uptake damper regulators, and in the case of forced draft, to the mechanism controlling the fan discharge. More simple systems, often employing a circulating fluid as the motive power instead of compressed air, act as overfire controllers, the fuel feed rate being proportioned to load by electric devices.

In the third method of draft control, electric, the most usual application takes the form of a small electric motor, low voltage, linked to the uptake damper, and causing the damper to assume a position at all times that will maintain a fixed and predetermined furnace draft intensity. The barometric and electric systems do not lend themselves to differential control, but the pneumatic and hydraulic are readily tied in with governors that will adjust the draft differential through the boiler in accordance with firing rate.

Oil Storage

About the last major topic facing the designer of an oil fired boiler plant is the layout of oil storage. The storage capacity is largely determined by the load and the availability of supply. In a large metropolitan area, deliveries are not a problem and storage can be more moderate than in the case of plants situated in rural areas where transportation means are less abundant. A week's supply at maximum firing is the smallest feasible storage under any conditions.

It is well to have at least two storage tanks per boiler room, to permit of cleaning or otherwise working on one while the other remains in service. The use of transport trucks for bulk oil deliveries from refineries or distributing points is now so common that the possibility should not be overlooked of piping for both rail unloading and truck shipments. If the tanks cannot be installed at a suitable level below the tracks to assure easy flow by gravity, it will be necessary to include unloading pumps in the system. Transport trucks of course are equipped with pumps for unloading. For rail deliveries, if the gravity unloading head is not available, the unloading pump should have a minimum capacity of 75 gpm, and if possible closer to 100 gpm. This means that the unloading of a 10,000 gal car can be accomplished in 1⅔ to 2¼ hr. All cars used in shipping heavy fuel oils are equipped with steam coils, but the customer must bring steam to the car.

Speed Defense

The principles discussed here have permitted rather novel handling of certain defense construction at remote points. Complete boiler-burner units were assembled, consisting of internally fired and therefore selfcontained boilers, mounted on suitable steel framework and equipped with induced draft fan, oil burner with combination full automatic and variable fire control, electric and steam oil heaters, feedwater heater, condensate return system, and the necessary interconnecting piping and wiring.

These units were completely fabricated, tested, crated and shipped to distant points, unloaded, set upon prepared foundations, hooked up to cold water, steam header, return header, suction and return fuel oil lines, electric power service, given a short discharge duct from fan outlet through roof, and were ready to start generating high pressure steam with complete regulation and absolute safety. No refractory settings were involved and no chimney had to be erected. For lack of ordinary bottled gas supply, acetylene tanks were requisitioned from the welding department.

Improved ventilation in mines was one of 10 recommendations made by the war production committee of the International Union of Mine, Mill and Smelter Workers in connection with operation of copper, zinc and lead mines on a seven day, three shift basis.

PLANTS INSTALL VENTILATION FOR WARTIME BLACKING OUT

Leo Hungerford Describes Some of the Methods Used

Evaporative air coolers at this aircraft plant provide comfortable working conditions in hot weather

BLACKOUTS of industrial plants for protection against possible air raids make imperative in many cases the installation of additional ventilation equipment to permit uninterrupted production for victory. Many plants—completely air conditioned—solve the problem by windowless construction. . . . Leo Hungerford, engineer, Utility Fan Corp., briefly describes here some of the other methods adopted on the outbreak of war

THE immediate necessity on the outbreak of war of blacking out Pacific Coast defense plants operating 24 hr a day made it imperative in many cases to provide additional ventilation. These systems must provide outside air during periods when windows are closed and covered, and must be so protected that they do not permit any light to leak out. The principal consideration in most cases has been speed of installation, which made it advisable to use quickly available equipment.

A number of the installations have been handled by one of the three following methods:

The first system includes placing double width, double inlet, ½ hp, light duty blowers, with bottom horizontal discharge, at intervals along one side of the building, discharging through the upper sections of the windows. Blowers and motors are enclosed in plywood cabinets, open at the bottom, to make them lightproof. On the opposite side of the building, lightproof louvers are placed in the upper sections of the windows. This permits cross-ventilation that can be controlled according to the outside temperature, by operating all or any of the blowers.

In the second method, used for buildings of larger area, ¾ hp double width, double inlet blowers are placed at intervals in the vertical sections of a saw toothed building or against the roof of a monitor type building. They discharge through window sections into a gooseneck duct, which makes the installation lightproof and watertight. Lightproof louvers are placed in the upper sections of the windows on three or four sides for supply air. This method is specially desirable where one section of the building has considerable amounts of dust and smoke, while another section is comparatively clean. It is also possible to vary the amount of ventilation according to the outside temperature by operating any desired number of blowers.

Cooling is combined with ventilation in the third method. Many companies desire air cooling to provide comfortable working conditions during hot weather. Evaporative coolers are placed at intervals on the roof to supply air through ducts into four-way plaques on the ceiling. For exhausting, light duty blowers are placed at intervals along the outside walls, discharging through the upper sections of the windows. These blowers are in plywood cabinets, open only at the bottom. By this method, the amount of air supplied or exhausted can be controlled. It also has the advantage of supplying fresh, cool, filtered air for comfortable working conditions in hot weather.

Special conditions to be considered in the ventilation of blacked-out plants include the possibilities of gas attacks, sabotage or direct bomb hits. These may have some effect on the placement of equipment or the design of the system installed. For instance, the installation of the outside air intake on the roof should make it possible to give some ventilation during gas attacks, by running the supply blowers and closing the lower exhausts.

PRACTICAL PIPING PROBLEMS

W. H. Wilson Tells of Experiences With Piping in Service Since 1910

WE learn by experience—good and bad. For this reason, I think other readers may be interested in some experiences with a low pressure steam main, part of which has been in daily use at our plant since it was installed over 30 years ago.

The first section of the line is in the basement of the power plant building. This section of the pipe line is 30 in. diameter, has flanged joints, and is supported by flanged tees with base supports. Top inlets receive the exhaust steam from three air compressors, each with a 16 in. discharge, and one compressor with an 18 in. discharge. There are also two steam turbines each operating non-condensing, with a 20 in. exhaust steam connection to the 30 in. main. There are also exhaust steam connections from pumps and engines.

PRACTICAL PIPING design, installation, operation and maintenance involve many factors not found in the books. Some of these are brought out in this "biography" of a low pressure steam line, which has been in almost daily use at an industrial plant since its installation over 30 years ago. . . . Discharging condensate from a high pressure line into the low pressure main, condition of the inside of the piping and the proper method of turning on the steam in a pipe line of this kind are among the topics treated by Mr. Wilson, a member of HPAC's board of consulting & contributing editors

This section of the piping was installed in 1910, including an extension of the same size on one end which changes in direction and continues in reducing sizes for supply lines to a number of large buildings for the heating systems. The section of the steam piping in the power plant, outside of an annual shutdown of a few days for necessary repairs and maintenance work, has been in daily use from the time it was first installed.

This 30 in. pipe is of the spiral riveted type, steel, galvanized, wall thickness 3/16 in., rivets of 3/8 in. diameter, and width of spiral steel sheets 13 in. center to center of rivets. Distance between the flanged joints in this steam line is 20 ft.

Several steam traps from a high pressure steam main which is situated near this low pressure main discharge the condensate into the 30 in. main through a common discharge line that enters the main from the top, is continued through inside, and is bent 90 deg. with an enlarged open end to divert the condensation in a horizontal direction, instead of a straight down discharge, which would in time cause a hole to be worn through the lower part of the main. (Previous experience had demonstrated this; at another location in the plant a feedwater heater with a water distributing pan near the top, had a steam trap discharge line entering from the top of the heater and discharging directly downward into the pan. This discharge water caused a hole to be worn through the bottom of the pan.)

Several years ago it was necessary to install a 10 in. steam inlet connection on this main, which was done by electric welding a steel saddle flange on the top of the pipe. As the main was out of service, it was a favorable opportunity to see the inside of the pipe line.

It was expected that because of steam cylinder oil lubrication from the engines of the air compressors and other equipment that the interior surface of the piping would have a coating of residue from the oil used. On the contrary, the inside of the galvanized piping was clean and a visual inspection showed no trace of oil and no accumulation of rust scale or foreign matter.

There is a 30 in. gate valve in the main at one end of the power plant building. It is motor operated and controls steam mains that supply the heating systems of a number of buildings. During weather of moderate temperature it is not necessary to keep the heat on continuously, and the steam can be easily and quickly controlled by operating this valve. When the valve is closed, more low pressure exhaust steam goes in a mixed pressure steam turbine and helps to furnish power for plant operation.

The steam line on the opposite end of the power house is reduced to 24 in. in size and, with a 90 deg change in direction, extends 400 ft. It is anchored at each end, and with a double travel expansion joint anchored in the center. This section of the pipe line is of wrought steel pipe, 3/8 in. thick, oxyacetylene welded joints, and was installed in 1918.

The steam, at 3 psi pressure, was turned on this line for a test before the covering was applied. The usual practice of turning on steam slowly was followed and a leak was found in the lower part of a welded joint. This was first attributed to poor welding. The leak was taken care of by the welder and steam turned on for another test, and another joint was found leaking. This was rewelded and a test made.

I was in the tunnel during the test and followed along the line with the travel of the steam. The top of the pipe was hot for over a hundred feet in advance of the bottom. This caused the pipe line to rise from the roller pipe supports 1 to 2 in., and it then broke with an audible report in a welded joint.

The joint was rewelded, and a blind flange was removed from the extreme end of the pipe line. The steam was turned on as fast as the valve could be opened manually, heating the main the entire length. The steam was shut off, the blind flange replaced, and steam was turned on while the main was hot, left on, and the covering applied. Large size drain valves were installed and the few times it has been necessary to turn the steam on it has been turned on quickly without causing any trouble.

Heating · Piping · Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

More Engineers Needed to Win War

It is becoming more and more important for engineers to learn about the opportunities for employment with the Government, as well as in industry, according to an announcement of the U. S. Civil Service Commission, which requests the cooperation of engineering organizations in publicizing the Government's personnel needs.

For the **Win the War** program everyone needs to determine where he can best serve. Engineers with the necessary qualifications, who may be available because of priorities, should obtain Announcement No. 173 issued by the United States Civil Service Commission, Washington, D. C.

• • •

Coming in March

Factors Influencing the Heat Output of Radiators, by A. C. Davis, W. M. Sawdon and David Dropkin.

Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend.

Complete Report of the Proceedings of the 48th Annual Meeting of the ASHVE.

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Vapor Pressure of Ice From 32 to -280 F

By John A. Goff*, Philadelphia, Pa.

Second progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Towne Scientific School, University of Pennsylvania

SUMMARY—The vapor pressure of ice for the range 32 to -280 F has been calculated from most reliable information on specific enthalpies and specific volumes now available. Final results are exhibited in the form of tables, one of which gives values intended to supersede data now listed in THE GUIDE

Introduction

VALUES of the vapor pressure of ice for the range 0 to —90 C were published by E. W. Washburn (1)† in 1924 and later included in International Critical Tables (2). These values have since been generally accepted as sufficiently accurate except possibly for applications to low temperature humidity control (3). Recently, however, important revisions in the fundamental data on which Washburn based his calculations have appeared making it desirable to recalculate the vapor pressure and extend the range to lower temperatures.

The task of recalculating the vapor pressure of ice was undertaken as a part of the cooperative investigation between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania. Since this investigation has for its ultimate objective a formulation of the thermodynamic properties of moist air which, on account of accuracy and thermodynamic consistency, can claim universal acceptance as standard, and since values of the vapor pressure of ice are necessarily a part of such a formulation, it seems proper to offer the present paper as a second progress report of the investigation.

The first progress report (4), which was previously presented to the Society, gave preliminary results on the experimental measurement of the interaction constant for moist air at a single temperature of 15 C. Owing to the desirability of making some improvements in the apparatus before attempting to obtain final results covering as wide a range of temperature as possible, no additional experimental data can be presented at this time.

The Clausius-Clapeyron Relation

This identical relation of thermodynamics provides the basis for calculating vapor pressure from other known data. It is merely a mathematical statement of the conditions for co-existence of two distinct phases of a pure substance in the absence of capillary and all other external forces except pressure, as explained in chapter 1, THE GUIDE 1942 (5). These conditions are equality between the two co-existing phases of (*a*) pressure, (*b*) temperature, (*c*) specific free enthalpy. The relation may be regarded as exact.

If h_i, v_i denote the specific enthalpy and specific volume of the saturated solid (ice) at absolute temperature T and if h_g, v_g denote specific enthalpy and specific volume of the saturated vapor at the same temperature T, then the Clausius-Clapeyron relation requires that the saturation pressure p_s vary with temperature T in the following manner,

$$\frac{dp_s}{dT} = \frac{1}{T} \cdot \frac{h_g - h_i}{v_s - v_i} \quad \ldots\ldots (1)$$

Since this relation is exact, the range and accuracy of values of vapor pressure calculated from it are limited only by the range and accuracy of available knowledge regarding the quantities appearing in its righthand member, including the absolute temperature T.

The Specific Enthalpy of the Saturated Vapor, h_g

As explained in Chapter 1 of THE GUIDE 1942 (5), this differs from the zero-pressure specific enthalpy h°_w by a small amount which, at a given temperature, is proportional to the pressure; thus,

$$h_g = h^\circ_w - B_{ww} \cdot p_s \quad \ldots\ldots (2)$$

In the present problem it is to be understood that p_s denotes saturation pressure. Values of the zero-pressure specific enthalpy h°_w are obtained from spectroscopic measurements. Stephenson and McMahon (6) have calculated values of the zero-pressure rotational specific heat of *normal* water vapor from 5 to 300 K. *Normal* water vapor is the 3:1 mixture of the *ortho* and *para* modifications in stable equilibrium only at temperatures above 100 K, but always observed, under ordinary conditions, at all accessible temperatures below this. Their results determine the magnitude of the distortion effect due to centrifugal forces which must be added to the values of specific heat calculated by A. R. Gordon (7) for temperatures above 300 K. From these two sources can be obtained reliable values of the total zero-pressure specific heat of normal water vapor clear down to 5 K and, by integration, corresponding values of zero-pressure specific enthalpy h°_w. Best values are listed in the column (1), Table 1, where 32 F has been taken as reference point.

Returning to Equation 2 the factor B_{ww} is a temperature function obtained by differentiating the second virial coefficient A_{ww} (5) according to the definition,

$$B_{ww} = -T^2 \frac{d(A_{ww}/T)}{dT} \quad \ldots (3)$$

Values of this quantity are obtainable from the equation of Keyes (8); and while there may be some question regarding the validity of this equation at the low tempera-

*Dean, Towne Scientific School, University of Pennsylvania. MEMBER of ASHVE.

†Numbers in parentheses refer to the bibliography at end of paper.

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Table 1—Relative Magnitudes of Terms in Equation 10

t	1 h°_w Btu per Lb	2 B_{ww}/RT	3 Δh Btu per Lb	4 h_l Btu per Lb	5 A_{ww}/RT	6 $1218.68\left(\frac{1}{T_0}-\frac{1}{T}\right)$ Btu Per Lb	7 $\int_{T_0}^{T}\frac{(h^{\circ}_w-h_l)\,dT}{T^2}$	8 $I\,(p_s, T)$	9 $\log_{10}\left(\frac{p_s}{p_0}\right)$
32	0.0	0.615	0.0452	0.0	0.0901	0.0	0.0	0.0	0.0
0	— 14.199	1.032	0.0455	— 15.586	0.1442	—0.172530	—0.000112	—0.000041	$\bar{1}$.31968
— 32	— 28.379	1.812	0.0456	— 30.171	0.2442	—0.370877	—0.000385	—0.000056	$\bar{2}$.53713
— 64	— 42.542	.....	0.0457	— 43.769	.....	—0.601304	—0.000684	—0.000059	$\bar{3}$.62811
— 96	— 56.695	.....	0.0458	— 56.361	.....	—0.872280	—0.000797	—0.000059	$\bar{4}$.56012
—128	— 70.841	.....	0.0459	— 67.962	.....	—1.195539	—0.000382	—0.000059	$\bar{5}$.28822
—160	— 84.982	.....	0.0460	— 78.630	.....	—1.587829	+0.001099	—0.000059	$\bar{7}$.74856
—192	— 99.119	.....	0.0461	— 88.363	.....	—2.073906	+0.004514	—0.000059	$\bar{9}$.84703
—224	—113.251	.....	0.0462	— 97.202	.....	—2.691967	+0.011332	—0.000059	$\bar{11}$.43893
—256	—127.381	.....	0.0463	—105.174	.....	—3.504216	+0.024131	—0.000059	$\bar{14}$.28935
—288	—141.509	.....	0.0465	—112.207	.....	—4.619225	+0.047804	—0.000059	$\bar{19}$.98984

tures considered in the present problem, it will be shown that the uncertainty introduced into the final results is very slight because the vapor pressure is so small. A few of the computed values of B_{ww}/RT are listed in column (2) Table 1.

The Specific Enthalpy of the Saturated Solid, h_i

Giauque and Stout (9) have measured the specific heat of ice at atmospheric pressure down to 15 K. By integration the corresponding values of specific enthalpy are easily derived. Strictly speaking these values should be corrected to saturation pressure p_s. For this purpose the identical relation,

$$\frac{\partial h}{\partial p} = v - T\frac{\partial v}{\partial T}, \quad \ldots\ldots\ldots (4)$$

may be used; and it is probably correct to assume that the righthand member is independent of pressure so that (4) may be applied in the form

$$\Delta h = \left[v - T\frac{\partial v}{\partial T}\right](P - p_s), \quad \ldots\ldots (5)$$

P denoting atmospheric pressure.

Measurements of the linear coefficient of expansion β by Jakob and Erk (10), extending almost to 20 K, are available for estimating the magnitude of the bracketed term in (5). Multiplied by three, these data give the volume coefficient of expansion α defined as follows:

$$\alpha = \frac{1}{v_0}\frac{\partial v}{\partial T} \quad \ldots\ldots\ldots\ldots (6)$$

where v_0 denotes the specific volume at 0 C. Thus, (5) becomes

$$\Delta h = v_0[1 + \int_0^t \alpha \cdot dT - \alpha T]\,(P - p_s) \ldots\ldots (7)$$

The necessary integration is easily performed, and values of Δh are listed in column (3), Table 1. It is to be noticed that, although the absolute magnitude of Δh is considerable, nevertheless its variation with temperature, which is the only thing of importance in the final results, is entirely negligible. Consequently this correction may be ignored and the values of specific enthalpy derived from the specific heat data of Giauque and Stout (9) may be regarded as directly applicable to the saturated solid. The results are given in column (4), Table 1, the reference point being 32 F as before.

The Latent Heats of Sublimation at 32 F

In order to relate the reference-point values of h°_w and h_l given in Table 1, it is necessary to know the latent heat of sublimation at 32 F and, in addition, the magnitude of the term $B_{ww}p$ at 32 F. Osborne (11) has recalculated earlier values of the latent heat of fusion and given 333.48 ± 0.2 international joules per gram as the best value. Osborne, Stimson and Ginnings (12) have published recent determinations of the latent heat of vaporization and of the vapor pressure at 32 F and have given as best values, 2500.17 international joules per gram, 0.006228 kg per square centimeter. The magnitude of the term $B_{ww}p_s$ is readily computed from the information referred to in the section on specific enthalpy of the saturated vapor and is 0.208 Btu per pound. From these data, the best value of the difference between h°_w and h_l at 32 F is determined to be 1218.68 Btu per pound.

Examination of Equation 1 will show that if specific enthalpies are measured in international electrical units while pressures and specific volumes are measured in absolute units, it is important to know the relation between the two units. The value 1.00019 absolute joule per international joule recommended by Osborne, Stimson and Ginnings (12) has been used; and this determines the conversion factor

$$1 \text{ Btu} = 778.18 \text{ ft lb} \ldots\ldots\ldots\ldots\ldots (8)$$

The Specific Volume of the Saturated Vapor, v_g

As explained in The Guide 1942 (5) the specific volume of the saturated vapor may be calculated from

$$p_s v_g = RT - A_{ww}p_s \ldots\ldots\ldots\ldots (9)$$

where A_{ww} is the second virial coefficient referred to in Equation 3. A few of the computed values of A_{ww}/RT are listed in column (5), Table 1. The value of R used is

$$R = 0.110236 \text{ Btu/lb F.}$$

The specific volume of the saturated solid v_i is at most 0.000006 v_g and therefore entirely negligible so far as application of (1) is concerned.

Arrangement for Computation

Dividing both sides of the Clausius-Clapeyron equation by pressure p_s and incorporating the results discussed in preceding paragraphs, this equation may be rewritten as follows

$$\frac{d\log_e p_s}{dT} = \frac{1218.68 + h^{\circ}_w - h_l - B_{ww}\cdot p_s}{RT^2\left(1 - \frac{A_{ww}}{RT}\cdot p_s\right)}$$

Rearranging somewhat and integrating, the final equation is

$$R\log_e\left(\frac{p_s}{p_o}\right) = 1218.68\left(\frac{1}{T_o} - \frac{1}{T}\right) + \int_{T_o}^{T} \frac{(h^\circ_w - h_1)}{T^2}\,dT + I(p_s, T) \quad .(10)$$

where

$$I(p_s,T) = \int_{T_o}^{T} \left\{ \frac{\frac{A_{ww}p_s}{RT}(1218.68 + h^\circ_w - h_1) - B_{ww}p_s}{T^2\left(1 - \frac{A_{ww}p_s}{RT}\right)} \right\} dT \quad \ldots\ldots\ldots \ldots(11)$$

In Equation 10, p_o is the vapor pressure at 32 F ($T_o = 491.70$). At present the value of p_o is only given to four significant figures (0.006228 kg per square centimeter) so that values of p itself calculated from Equation 10 are only good to four significant figures. Moreover, the calculated values of $\log_{10}(p_s/p_o)$ are subject to the uncertainty in the heat of sublimation at 32 F which is probably of the order of 2 or 3 parts in 10,000 (13); also to the uncertainty in the absolute temperature corresponding to 32 F which is probably 1 part in 10,000. Therefore the values listed in Table 3 are not to be regarded as accurate beyond the fourth significant figure with slight uncertainty in the fourth significant figure at the lower temperatures.

The solution of Equation 10 is effected in the following manner: (*a*) provisional values of p_s are calculated ignoring the term $I(p_s, T)$; (*b*) these provisional values are then used to evaluate $I(p_s, T)$; (*c*) since $I(p_s, T)$ is so very small no revision is necessary and final values of p_s/p_o are obtained directly. The relative magnitudes of the three terms in Equation 10 are shown by the figures in columns (6, 7, 8) Table 1. In this table 32-deg intervals are shown, but in the actual computations 8-deg intervals were used, the integrals being evaluated by mechanical quadrature applying the Gregory formula (14).

Final Results

Final results are exhibited in two tables. The first, Table 2, lists values of $\log_{10}(p_s/p_o)$ at 8-deg intervals of temperature. Values at intermediate temperatures can easily be obtained by constructing a table of differences and applying Bessels' formula for interpolation to halves (14).

In Table 3 are given values of the vapor pressure of ice from 32 to —60 F at 1-deg intervals, the unit of pressure being the inch of mercury defined as follows:
1 in. Hg (32 F) = 0.033421 standard atmospheres = 0.034532 kg per square centimeter.

Table 2—Values in Equation 10 at 8 Deg Intervals

t	$\log_{10}\left(\frac{p_s}{p_o}\right)$		$\log_{10}\left(\frac{p_s}{p_o}\right)$		$\log_{10}\left(\frac{p_s}{p_o}\right)$	t	$\log_{10}\left(\frac{p_s}{p_o}\right)$
32	0.0	— 48	$\bar{2}.10023$	—128	$\bar{3}.28822$	—208	$\bar{10}.71819$
24	$\bar{1}.83842$	— 56	$\bar{3}.86884$	—136	$\bar{6}.93145$	—216	$\bar{10}.09919$
16	$\bar{1}.67136$	— 64	$\bar{3}.62811$	—144	$\bar{6}.55685$	—224	$\bar{11}.43893$
8	$\bar{1}.49855$	— 72	$\bar{3}.37753$	—152	$\bar{6}.16306$	—232	$\bar{12}.73311$
0	$\bar{1}.31968$	— 80	$\bar{3}.11645$	—160	$\bar{7}.74856$	—240	$\bar{13}.97681$
— 8	$\bar{1}.13446$	— 88	$\bar{4}.84423$	—168	$\bar{7}.31167$	—248	$\bar{13}.16440$
—16	$\bar{2}.94254$	— 96	$\bar{4}.56012$	—176	$\bar{8}.85055$	—256	$\bar{14}.28935$
—24	$\bar{2}.74355$	—104	$\bar{4}.26337$	—184	$\bar{8}.36311$	—264	$\bar{15}.34406$
—32	$\bar{2}.53713$	—112	$\bar{5}.95311$	—192	$\bar{9}.84703$	—272	$\bar{16}.31966$
—40	$\bar{2}.32283$	—120	$\bar{5}.62840$	—200	$\bar{9}.29970$	—280	$\bar{17}.20571$

Summary

The vapor pressure of ice for the range 32 to —280 F has been calculated from most reliable information on specific enthalpies and specific volumes now available. The range could have been extended to lower temperatures since all necessary information is available down to at least —400 F; but it was felt that the range 32 to —280 F will supply every practical need. The data upon which the calculations are made are characterized by a high degree of accuracy and permanence; and, since the basis of calculation is one of the identical relations of thermodynamics, namely, the Clausius-Clapeyron equation, it is felt that the final results also possess a high degree of accuracy and permanence.

The final results are exhibited in the form of tables. Table 2 lists values of $\log_{10}(p_s/p_o)$, where p_o denotes the vapor pressure at 32 F, at 8-deg intervals. By constructing a corresponding table of differences these data can easily be interpolated by application of Bessels' Formula for interpolation to halves. Table 3 lists values of the vapor pressure itself for the range 32 to —60 F at 1-deg intervals which are intended to supersede the values listed in Table 6, THE GUIDE 1942.

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Table 3—Vapor Pressure of Ice

t (F)	$p_s \times 10$ (IN. HG)	t (F)	$p_s \times 10$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)
....	.....	29	1.5710	19	9.7904	9	5.9788	0	3.7655	—10	2.2042	—20	1.2591	—30	7.0081	—40	3.7929	—50	1.9923
....	.....	28	1.4997	18	9.3280	8	5.6845	—1	3.5729	—11	2.0866	—21	1.1888	—31	6.5993	—41	3.5613	—51	1.8649
....	.....	27	1.4314	17	8.8857	7	5.4034	—2	3.3894	—12	1.9747	—22	1.1223	—32	6.2125	—42	3.3428	—52	1.7451
....	.....	26	1.3660	16	8.4626	6	5.1351	—3	3.2146	—13	1.8684	—23	1.0592	—33	5.8468	—43	3.1367	—53	1.6324
....	.....	25	1.3033	15	8.0579	5	4.8791	—4	3.0481	—14	1.7673	—24	0.99929	—34	5.5011	—44	2.9426	—54	1.5265
....	.....	24	1.2432	14	7.6708	4	4.6348	—5	2.8895	—15	1.6713	—25	0.94254	—35	5.1743	—45	2.7596	—55	1.4270
....	.....	23	1.1857	13	7.3009	3	4.4017	—6	2.7385	—16	1.5801	—26	0.88879	—36	4.8655	—46	2.5872	—56	1.3335
32	1.8036	22	1.1306	12	6.9474	2	4.1794	—7	2.5948	—17	1.4935	—27	0.83788	—37	4.5738	—47	2.4248	—57	1.2457
31	1.7228	21	1.0779	11	6.6097	1	3.9675	—8	2.4581	—18	1.4113	—28	0.78966	—38	4.2984	—48	2.2718	—58	1.1633
30	1.6453	20	1.0274	10	6.2870	0	3.7655	—9	2.3280	—19	1.3332	—29	0.74401	—39	4.0383	—49	2.1278	—59	1.0860
																		—60	1.0135

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Twentieth Anniversary Edition

THE 1942 edition of HEATING VENTILATING AIR CONDITIONING GUIDE marks the Twentieth Anniversary of publication—1922-1942. In 1922, the first issue of THE GUIDE was a book of only 360 pages, limited in its coverage of the technical phases of the industry and in presentation of manufacturers' equipment data. The 1942 edition contains 1,256 pages and treats exhaustively every phase of heating, ventilating and air conditioning, and related phases of refrigeration.

During the 20 years of publication, many men of long training and expert knowledge of their respective fields have contributed engineering data and general information that have made the Technical Data Section the accepted authority in every branch of the profession and the industry. THE GUIDE is now widely used as a reference book in America and abroad; in many schools and colleges it is used as a text and reference book.

In this new 1942 edition a new chapter has been added—Chapter 3, on Fundamentals of Heat Transfer—which includes the basic equations for conduction, convection and radiation. A detailed solution is given for a problem involving all three mechanisms of heat transfer. Some revisions have been made in the chapter on Thermodynamics of Air and Water Mixtures. The Mollier Diagram for Moist Air has been redrawn, and a new Volume Diagram for Moist Air has been added.

The data on Physiological Principles have been revised to include current knowledge on thermal interchanges taking place between the human body and the environment. The chapter on Central Systems for Comfort Air Conditioning has been

completely rewritten. Revisions of the chapters on Air Distribution, Air Duct Design, Sound Control and Fans, provide new data on the design of air handling systems. A new air friction chart, developed at the ASHVE Research Laboratory, is included in the chapter on Air Duct Design.

The chapter on Radiant Heating has undergone major revisions and contains data essential for the design of radiant heating systems. A new chart has been added to the chapter on Pipe and Duct Heat Losses. A group of refrigerating graphical symbols for drawings has been added to the chapter on Terminology. In addition, other chapters have been reviewed and revised.

A detailed index and many footnotes and cross references facilitate the fullest use of the data contained in this section of the book.

The Catalog Data Section, too, has grown during the 20 years of publication. It has again been revised and gives factual data on the latest types of equipment. Leading manufacturers of heating, ventilating and air conditioning apparatus and materials present their products in 5 sub-divisions, profusely illustrated and described. Much detailed information is given, including sizes and capacities, which enables GUIDE users to determine the suitability of equipment for specific uses. These data are carefully listed in an Index to Modern Equipment.

On the title pages of each subdivision are many references to the Technical Data Section, thus coordinating the two sections of THE GUIDE and enabling the reader to obtain complete information on a specific subject.

In the earlier years of publication, THE GUIDE was distributed chiefly among the members of the Society and other organizations interested in this field. With each issue the field has broadened and distribution has increased and THE GUIDE is now used by more than 10,000 men annually. These men are located in every part of the United States and Canada, and in many foreign countries, serving the profession and the industry and giving THE GUIDE world-wide prestige.

This year THE GUIDE is bound in a stiff blue cover with gold stamping and is now being distributed to the membership. Individual copies can be obtained at $5.00 per copy, and a thumb-indexed edition will be available with chapter tabs at $5.50 for those who desire this particular feature.

Panel Heating and Cooling Performance Studies

By B. F. Raber* and F. W. Hutchinson**, Berkeley, Calif.

First progress report of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of California.

SUMMARY—This paper gives experimental data for the feeling of warmth on 50 subjects in a reflecting enclosure with air at 60 F and surfaces at 57 F, and with no heating other than that due to the occupants. An investigation was made of surface emissivity and its practical significance in the design and operation of any type of heating system

Introduction

THIS project covers a particular part of the radiant transfer problem which is as important to convective as to panel installations. In an earlier paper,[1] the authors developed a rational approach to panel analysis and discussed, from theory, the effect of the surface characteristic, emissivity, as one term in the basic radiation equation. The present paper reports on an experimental investigation of this emissivity factor and its practical significance in the design and operation of any type of heating system. Except where otherwise stated, the analysis, experimental results and general conclusions which follow apply with equal validity to warm air, convector, cast-iron radiator, or panel systems.

Heating systems, whether convective or radiant, have as their objective the maintenance of that relationship between effective air temperature, ET, and mean radiant temperature, MRT, which corresponds to optimum comfort. Each system has an effect, direct or otherwise, on both variables.

The possibility of realizing comfort in a room with air and surfaces at a uniform temperature of 70 F is well established. Likewise, the possibility of attaining comfort with air at 60 F and surfaces at 80 F has been reported.[2] European experience with panel cooling indicates that comfort can also be expected when air is 80 F and surfaces 60 F. In this connection convective heating and panel cooling may be considered similar because each establishes comfort with air temperatures above 70 F and surface temperatures below 70 F; a like similarity holds between convective cooling and panel heating. The object of the present investigation was to determine the possibility of establishing comfort when both air and surface temperatures are below 70 F; experimental data are offered to substantiate the conclusion, valid from theory, that a room with both air and surfaces at 60 F can be comfortable, regardless of the type of heating system employed, if certain surface and size requirements are satisfied.

Principles

Within the limits of usual operating temperatures, conditions of comfort exist when ET and MRT bear a relationship to one another such that the body's sensible heat loss remains practically constant. At reduced air temperature the convective fraction of sensible loss increases, and therefore a corresponding decrease in the net loss by radiation becomes necessary. Heating systems of the panel type are usually designed to effect a decrease in the net radiant loss by increasing the emissive power of the surrounding surfaces. The investigation covered by this paper deals with environmental systems in which a similar reduction of net radiant loss is accomplished by decreasing the effective absorptivity of enclosure surfaces and thereby reflecting back to the occupants some of the radiant energy which they emit.

As an example consider a room in which all surfaces are uniformly heated to 83 F (emissivity taken as 0.9) and the air temperature is at a value corresponding to comfort, approximately 57 F. The average surface temperature of the clothed occupants can be taken as 83 F[3] so there is no net radiant heat loss from the occupants. From an examination of this system one can readily conclude that, if the above conditions are comfortable, comfort will exist in any other environment for which the air temperature is 57 F and the net radiant loss is zero. Assume existence of a surfacing material having emissivity and transmissivity of zero (100 per cent reflectivity). Replace the 83 F enclosure with one having a 100 per cent reflective inside surface. Now, *regardless of the surface temperature*, there can be no net radiant transfer to or from the occupants and the flux in the enclosure must increase until the radiant energy being returned to the occupants after n reflections is equal to that which would be received if the enclosure were at a uniform surface temperature equal to that of the occupants, 83 F; conditions with respect to comfort are then the same as existed

*Professor of Mechanical Engineering, University of California. Member of ASHVE.
**Assistant Professor of Mechanical Engineering, University of California.

[1]Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson, (ASHVE Journal Section, *Heating Piping and Air Conditioning*, August 1941, p. 512.)

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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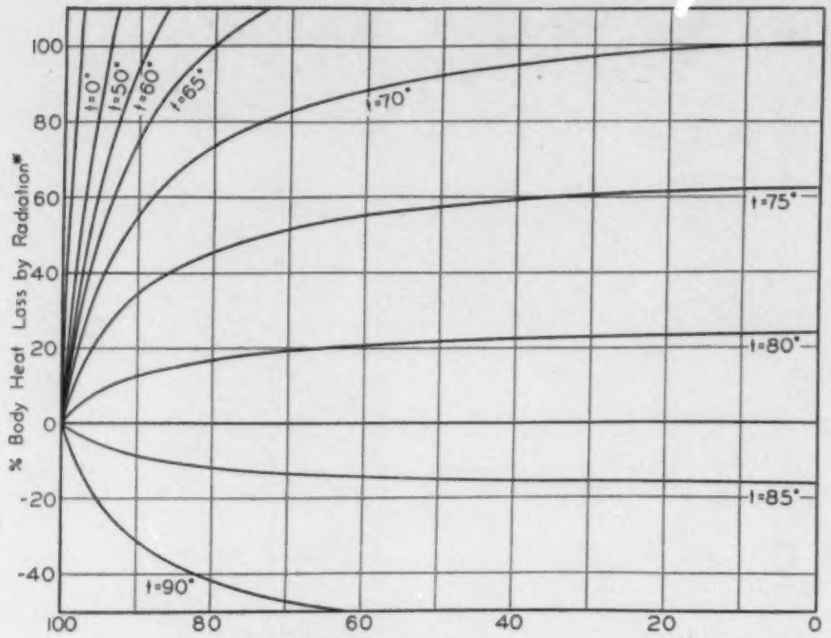

Fig. 1—Temperature-emissivity characteristics of a 150 sq ft enclosure
*The heat loss scale is expressed in per cent of that loss of body heat by radiation which occurs in a room with walls (e=1.0) at 70 F and air at 70 F.

in the original enclosure with surfaces heated to 83 F.

While the comfort characteristics of the two enclosures described above are identical, the heating requirements are vastly different. In the first case a temperature difference of 26 F exists between the surrounding surfaces and the inside air; heat loss to the ventilation air would therefore be high. In the second case, the only heat required would be that needed to raise the ventilation air to 57 F and to supply transmission losses for the air-to-air temperature difference; the enclosure surface would come to an equilibrium temperature less than 57 F and the requisite heat input to the system could be achieved with any kind of heating system. With perfectly reflecting surfaces the size and shape of the room would not affect the condition of zero net radiant transfer.

Non-Perfect Reflecting Surfaces

A perfectly reflecting surface, as assumed above, does not exist. In extending the principle of *insulating* the occupants against radiant heat loss an examination is desirable regarding the extent to which existing materials can be employed to reduce net loss without need of increasing the emissive power of the surrounding surfaces.

Effect of Reflectivity

Fig. 1 shows the net radiant transfer between the occupant and the surroundings expressed as a percentage of that transfer rate which exists when walls are at 70 F and the room is large with respect to the occupant (effective emissivity of enclosure surface is then unity). The curves give per cent transfer as a function of enclosure reflectivity, for a given surface temperature, and are calculated for an occupant having 15.5 sq ft of effective radiating surface at 83 F[4]. To avoid the necessity of specifying shape factors, yet to obtain greater accuracy than is possible with the usual assumption that *the enclosed body is small with respect to the enclosure,* the curves are plotted from the exact analytical equation for concentric spheres or for concentric cylinders of great length.[5] The enclosure area is taken as 150 sq ft, equal to the area of the test chamber used in the experimental work. Reflection was considered as from perfectly diffusing (matte) surfaces.

Drawing a horizontal line through the 100 per cent transfer point of Fig. 1, the intersections with temperature curves give the respective reflectivities at which the net transfer from or to the occupants would be equal to that with a black body enclosure at 70 F. Interpreted in terms of comfort these reflectivities must exist for surfaces of the corresponding temperature if a condition of comfort is to be realized equivalent to that which would obtain if black body surfaces at 70 F existed. From Fig. 1 equivalent comfort would be realized in a room with air at 70 F and any of the following surface temperature-reflectivity combinations:

[4]The ASHVE Guide, 1941, p. 761.
[5]*Heat Transmission*, W. H. McAdams, Case 5, p. 54.

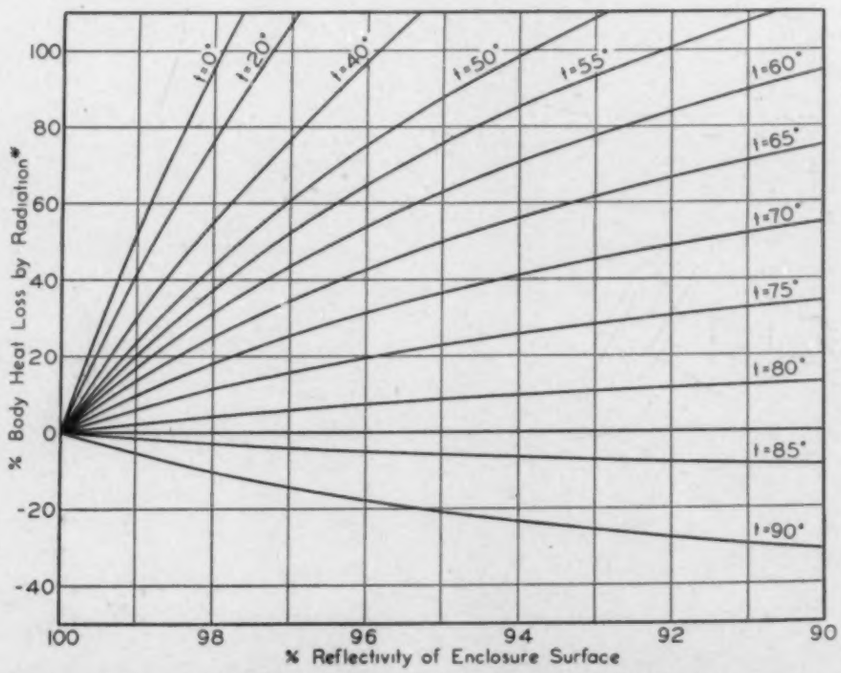

Fig. 2—Temperature-emissivity characteristics of Fig. 1 increased to larger scale

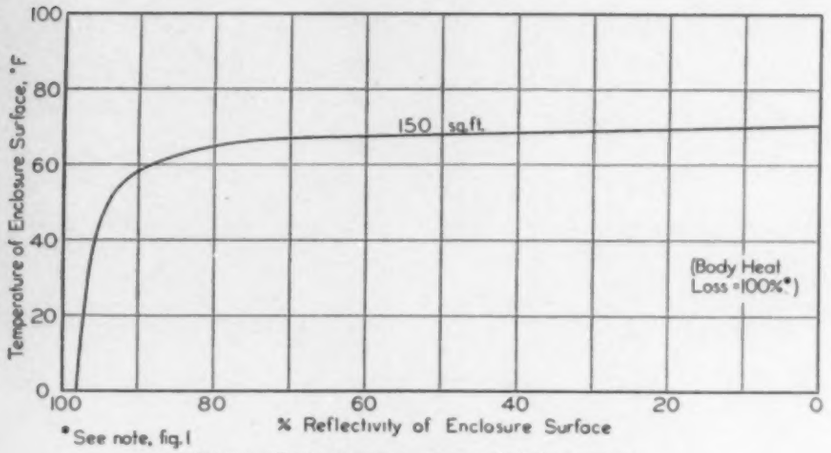

Fig. 3—Typical temperature-emissivity curves

Reflectivity Per Cent	Surface Temperature Fahrenheit
0	70
89	60
94	50
96	40
98	0
100	—460 to infinity

The significant area of Fig. 1 is crowded into such a small section of the chart that this area has been redrawn (Fig. 2) to a much larger scale.

The complete curve for temperature as a function of reflectivity at constant net radiant transfer of 100 per cent (as defined for Fig. 1) is shown in Fig. 3. From the preceding tabulation and Fig. 3 it is evident that reflectivity becomes of practical importance only when its value is in excess of 90 per cent. Between 90 per cent and 100 per cent a small change in reflectivity results in a large variation in net transfer and therefore has an important effect on the necessary surface temperature.

Diffuse vs. Specular Reflection

The assumption of diffuse (matte-like) reflection corresponds closely to the condition of actual surfaces. If, however, *regular* or *specular* (mirror-like) reflection could be realized, the effectiveness of the surface as a form of insulation against radiant loss would be increased. Fig. 4 compares net energy transfer as a function of reflectivity for two surfaces, both at 70 F, one of which is 100 per cent diffusing and the other a perfect specular reflector. An inspection of the figure shows that the characteristics of both surfaces are identical for reflectivities of 0 per cent or 100 per cent and that as these values are approached the characteristics also draw together. In the reflectivity range from 40 per cent to 95 per cent the specular surface has a definite advantage. The reflectivity corresponding to maximum difference between the transfer rates occurs at that value for which the slope of the diffuse reflection curve is parallel to the curve for specular reflection; for the case shown in Fig. 4 this maximum difference is reached at 78 per cent. Inspection shows that the advantage of a specular reflecting surface is slight in the range of high emissivities, but is of great significance in the lower emissivity range which corresponds to polished metal surfaces.

Effect of Room Size

Perfectly reflecting surfaces cannot exchange radiation with the occupants regardless of room size. Perfectly absorbing materials do not permit reflections and hence establish an energy transfer that is likewise independent of room size. All other surfaces, however, absorb a fraction of the energy leaving the occupants and return the remainder. The fraction re-absorbed by the occupants depends not only on the emissivities of surrounding surfaces, but also on the number of re-reflections that take place between enclosure surfaces for each reflection which causes energy to strike the occupants. As the room size increases it is therefore evident that fewer of the infinite number of energy paths through the room will strike the occupants, hence the fraction of energy re-absorbed will decrease. Fig. 5, plotted from the same equation as Fig. 1, gives curves for energy transfer as a function of reflectivity for three enclosure areas all at the same temperature. The areas for which the curves apply are 900 sq ft (corresponding to a room 8 ft 6 in. x 14 ft x 15 ft); 150 sq ft (the area of the experimental enclosure); 16.5 sq ft (the approximate inside area of clothing for an individual of average size). Fig. 6 gives enclosure area as a function of reflectivity for a fixed surface temperature of 70 F and a constant heat transfer one half (or 75 per cent) as great as that from an occupant to a 70 F *black body* enclosure.

The effect of room size on the performance of non-perfect reflecting surfaces is evident from Figs. 5 and 6. As the room becomes smaller

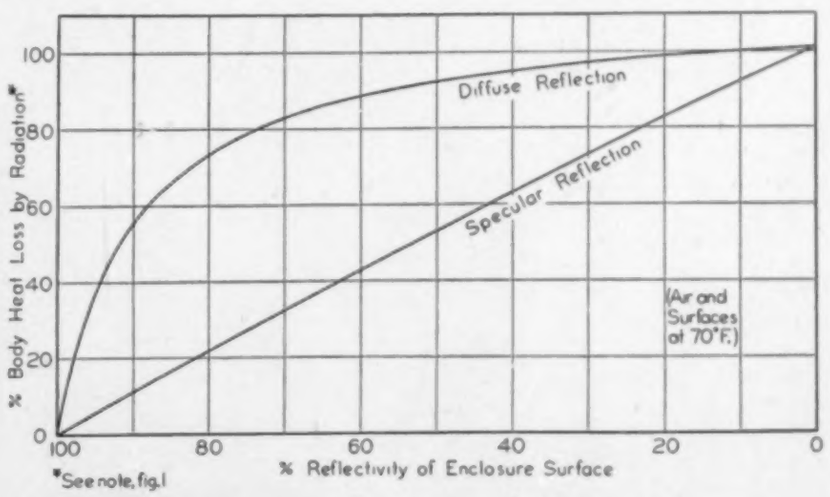

Fig. 4—Specular vs. diffuse reflection in 150 sq ft enclosure

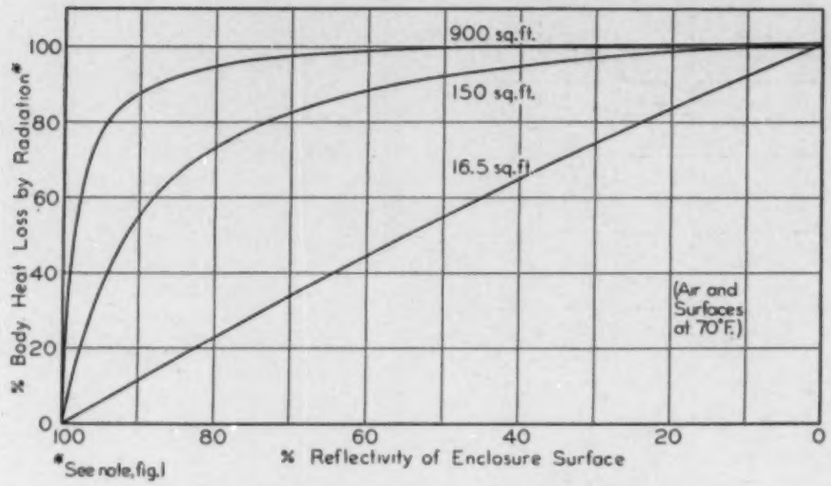

Fig. 5—Effect of room size on reflection characteristics

the equation expressing radiant transfer approaches the form used for exchange between infinite parallel planes; this latter equation is identical with that for the condition shown in Fig. 4 with specular reflection. Decrease in enclosure size therefore increases effectiveness by causing conditions in the space to more closely approach those which would exist if the surface were a specular reflector.

Numerical Example

The discussion and curves of the preceding section bring out three characteristics of any enclosure designed to reduce, by reflection, the net radiant heat loss of the occupants. These are:

1. The effectiveness of reflectors is not a lineal function of reflectivity, but increases very rapidly as conditions of perfect reflection are approached.
2. Effectiveness increases as the surface approaches specular reflection.
3. Effectiveness increases rapidly as the size of the room decreases.

Evidently, therefore, an enclosure intended for experiment on reflection should be small and finished on the inside with a polished surface of low emissivity.

Consider an enclosure 5 ft x 5 ft x 5 ft (150 sq ft of surface). If the reflectivity were 100 per cent, comfort of an occupant would be realized, without energy input other than his own, when air and surfaces are at approximately 57 F. Since a real surface cannot be a perfect reflector the requisite comfort air temperature in an actual system must be sufficiently above 57 F to reduce the convection loss by an amount equal to the radiant exchange between surface and occupant. Referring to Fig. 2, the radiant loss to a 57 F surface having an emissivity of 0.025 (reflectivity of 97.5 per cent) is 41 per cent of that lost to a black body enclosure at 70 F. Black body loss at 70 F is 160 Btu per hour[6] so the radiant loss resulting from non-perfect reflection is $0.41 \times 160 = 65$ Btu per hour. Since the total sensible heat loss must be constant (for comfort), this 65 Btu per hour must be deducted from the convective loss. As a first approximation the convective loss may be considered directly proportional to the temperature drop from a clothed body (83 F) to the ambient air; the necessary air temperature is then,

$$t = 83 - (300 - 65)\frac{83 - 57}{300} = 62.6 \text{ F.}$$

Thus in an enclosure 5 ft x 5 ft x 5 ft having perfectly diffusing surfaces of 97.5 per cent reflectivity, comfort conditions would be anticipated with air at 62.6 F and surfaces at 57 F.

If the surface of the above enclosure gave specular reflection, the radiant heat transfer would be 0.045 (interpolated from Fig. 4) $\times$ 160 $=$ 7 Btu per hour and the requisite air temperature would then be,

$$t = 83 - (300 - 7)\frac{83 - 57}{300} = 57.7 \text{ F.}$$

Since an actual surface is a limited specular reflector its performance is somewhere between those just investigated and one would expect that the requisite air temperature would be within the above limits. Approximately, then, an actual polished surface enclosure 5 ft x 5 ft x 5 ft with emissivity near 0.025 would be expected to provide substantial comfort in air at 60 F when surfaces are 57 F.

Experimental Set-Up

Tests conducted in a small room accentuate the effect of reflection and hence increase the accuracy of the data. For this reason the reflection chamber designed for this project was made as small as practicable (5 ft x 5 ft x 5 ft). If actual comfort data from such a chamber agreed reasonably well with theory the functional size effects noted in Fig. 6 could be readily used to extrapolate results to larger rooms.

Copper has a higher reflectivity than most commercially available materials so the chamber was constructed of 20 oz sheet copper with inside metal surface exposed and carefully polished. The exterior structural frame was made of ⅛ in.

[6] The ASHVE Guide, 1941, p. 65.

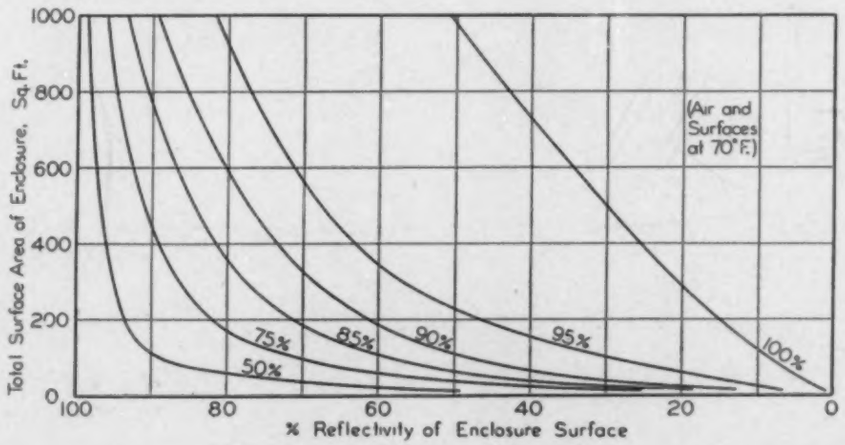

Fig. 6—Typical size vs. effectiveness curves

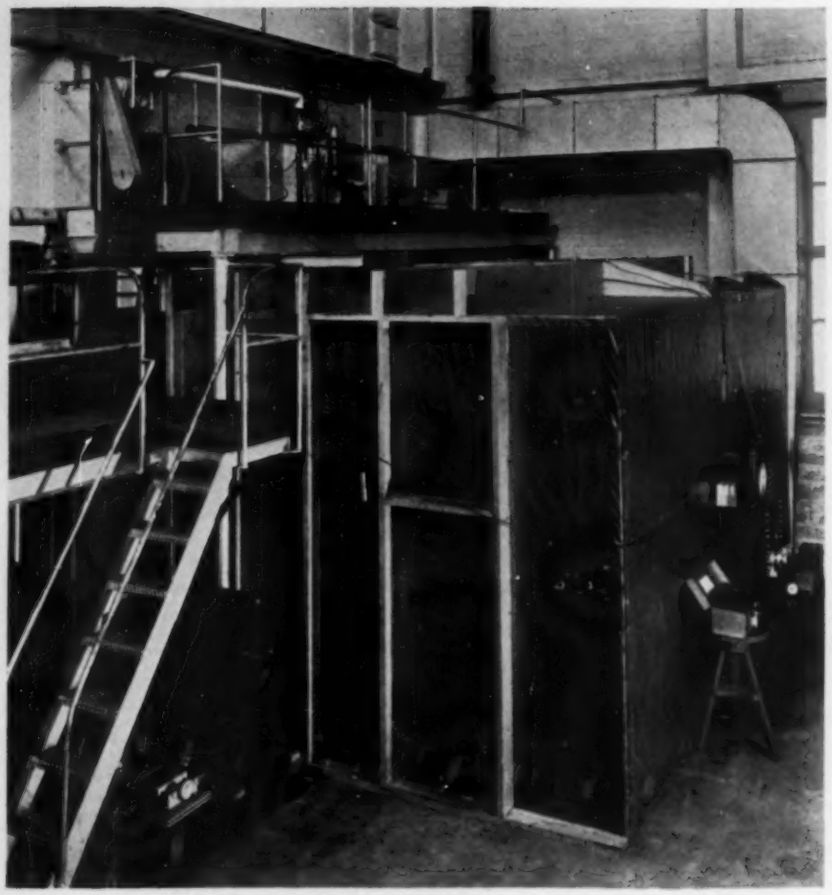

Fig. 7—Exterior view of air conditioning research rooms

x 1 in. x 1 in. angle iron to which the copper sheets were attached by means of polished No. 4-36 brass machine screws. Access to the chamber was through one of the sides, hinged at the top. Ventilation was provided through cracks equivalent in area to inlet and exhaust openings each 6 in. x 6 in. The occupant *saw* 149.5 sq ft of reflecting surface and 0.5 sq ft of crack. Taking the emissivity of the metal surface as 0.022 and the crack as a black body, the effective emissivity of the chamber was,

$$\frac{149.5 \times 0.022 + 0.5 \times 1.0}{150} = 0.025, \text{ or } 97.5$$

per cent reflectivity.

The actual test chamber therefore corresponded closely to the conditions of the numerical example already considered and, from theory, comfort would be expected with air at approximately 60 F and walls at 57 F.

The reflection chamber was placed on a raised platform inside one of the larger air conditioning research rooms, the exterior view of which is presented in Fig. 7. By means of equipment installed above the rooms the volume and temperature of the air passing through the larger room, and around the reflection chamber, could be controlled, and hence the surface of the reflection chamber or its contained air could be held at desired temperatures. The inside surface and the inside air temperatures are not independent variables, but for the chamber used in these experiments their relationship corresponded to the desired conditions. This can be shown, approximately, as follows:

Sensible heat transfer from a comfortable occupant to enclosure approaches 300 Btu per hour. The equivalent inside surface film coefficient of heat transfer (for emissivity of 0.025) approaches 0.8 Btu per hour per square foot per degree Fahrenheit.[7] Then,

$$q = hA\ (t_s - t_a)$$
$$300 = 0.8 \times 150\ (t_s - t_a)$$
$$t_s - t_a = \frac{300}{150 \times 0.8} = 2.5 \text{ F}$$

Thus for an inside air temperature of 60 F the surface temperature would be 57.5 F which corresponds closely with the desired value of 57 F. Experiment showed an equilibrium overall temperature difference from inside to outside air of 3½ F.

Thermocouples were placed on inside and outside surfaces of the reflection chamber and in the inside and ambient air. After considerable experiment the most satisfactory method of measuring inside air temperature was found to be through use of a copper-constantan thermocouple inserted in a drilled and sealed hole, edgeways in a piece of 3/32 in. x ½ in. x ½ in. copper; the metal was selected to be free of scratches or surface imperfections of any kind and was then carefully cleansed and polished. Readings of this thermocouple checked those from a double-shielded, aspirating, calibrated, dry-bulb thermometer.

Test Conditions

The sixty subjects tested were largely volunteers from the junior and senior classes in mechanical engineering. With a few exceptions the age limit was from 19 to 23; 56 of the subjects were male.

Adoption of a standard of clothing did not seem feasible because of the relatively large number of subjects. In consequence, subjects entered the test chamber in the same clothing which they had worn throughout the day. Of those tested 20 wore neither sweater nor coat and had shirts open at the neck; 21 wore sweaters of varying weights. The remaining 19 were dressed in two- or three-piece suits, or outside jackets, or, in two cases, wore both a sweater and a coat. While variation in the manner of dress was strikingly great it none the less represented the selected conditions of different subjects for comfort in the same outside atmosphere.

The only heat source in the room, other than the occupant, was a 6-watt globe which provided enough light to permit reading, or writing, during the period of occupancy. Humidity was not controlled, but

[7] Radiation and Convection from Surfaces in Various Positions, by G. P. Wilkes and C. M. F. Peterson. (ASHVE TRANSACTIONS, 1938, Vol. 44, Fig. 2, p. 517.)

in many of the tests the subject was asked to operate a small shielded-bulb aspirating psychrometer and to record the wet- and dry-bulb temperatures at the 30-min interval; the average relative humidity was 65 per cent. The maximum effect on comfort dry-bulb temperature of an increase in relative humidity amounting to 15 per cent greater than the 50 per cent standard would be an increase of 1 F; in the present tests, however, increased humidity would have an opposite effect of undetermined magnitude due to absorption by water vapor of radiant energy, with a consequent reduction in the amount of radiation being returned to the occupant. These two contrary effects of humidity are both small, so it was considered satisfactory to assume that they compensated for one another and therefore could both be neglected.

All tests were run between the hours of 11 a. m. and 5 p. m. on days when the air temperature outside of the air conditioning research room was between 70 F and 80 F; for over 75 per cent of the tests this outside temperature was between 73 F and 77 F. The effect of entering a cooled enclosure from an atmosphere of higher temperature is one of slight shock, so the warm outside conditions were considered conservative, since their effect, if any, would be to cause the occupant to be somewhat cooler.

The reflection chamber is exposed on all sides, top and bottom to the ambient air in the south air conditioning research room. To enter the chamber the subject was therefore obliged to pass through the ambient air which was approximately 3½ F lower (and at higher velocity) than the inside air in which each test was conducted. The average time of exposure to the ambient air was approximately 30 sec; in this interval the subject stepped into the enclosure and the side was returned to its closed position. As with the warm outside air, brief exposure to the colder ambient conditions should, if anything, have accentuated the shock and hence made the test results conservative.

A third acclimatizing factor existed in the necessity of entering an enclosure initially at a temperature 3½ F below that value assigned for the test. Since no heat was supplied to the enclosure its temperature at the start of each test was equal to that in the ambient space. However, the rise occurred rapidly following admission of the subject and within 4 min the air temperature was within 1 F of the desired value. Six minutes after occupancy started the air temperature had risen 3½ F and was at the value selected for test. The effect of this brief heating-up period would, again, be to slightly chill the subject and lead, therefore, to a conservative estimate on his part of the degree of comfort realized in the enclosure. Since the purpose of the tests was to investigate, experimentally, the effectiveness of reflection heating rather than to evaluate precise comfort reactions, it was considered satisfactory to neglect all such minor factors providing that, as in the three acclimatization effects just described, their omission would increase, rather than decrease, the difficulty of establishing comfort through reflection of body radiation.

Test Procedure

Instructions to Subjects

Before entering the reflection chamber each subject was asked to read the following instructions:

The purpose of the test which you are taking is to determine, statistically, the feeling of warmth experienced by an average individual when subjected to a particular type of environment. The physical data relative to conditions in the room during the interval in which you remain there are being recorded (from thermocouple readings) by the operator. After the test he will be glad to tell you the air temperature, wall surface temperature, etc. that existed during your period of occupancy.

The data which we ask that you record are subject to many variables and it may be that you will be unable to make a definite statement as to whether you are cool, warm, cold, hot, or ideally comfortable. In this event we ask that you record the best possible estimate.

Conditions in the room will be varied for different subjects so it is important that you do not enter with the preconceived idea that you *should* be comfortable, or cold, or hot. In some cases conditions will be changed during a test so that you may be uncomfortably warm at the start and cold before the test is over. Therefore you should record your general feeling of warmth at intervals of 5 min.

To establish a reasonable uniformity of reports you are asked to report your feeling of warmth, by number, from the following schedule:

1. Uncomfortably hot.
2. Uncomfortably warm.
3. Comfortably warm.
4. Ideally comfortable.
5. Comfortably cool.
6. Uncomfortably cool.
7. Uncomfortably cold.

The above schedule refers to your *overall* degree of comfort. In addition you may notice local conditions as drafts, heating or cooling of exposed skin, feet, back, etc.; after recording the number which defines your *over-all* feeling of warmth write any comments or observations on conditions which you consider unusual or significant.

After reading the instructions each subject was provided with a pencil, watch, and a standard form for recording his observations. The first reading was requested as soon as the reflection chamber door had been closed and subsequent readings at five minute intervals thereafter.

Position of Subject

The only furniture in the reflection enclosure was a small stool and two ¾ in. foot pads, of insulating material, arranged to be moved to that position on the floor for which the occupant was most rested. In entering the enclosure the subject was warned not to step on or touch with hands the reflecting surface. The subject was seated on the stool with feet resting in a comfortable position of his own selection on the foot pads.

Throughout the test the occupant was obliged to remain in a sitting position and was not free to alter his position other than through slight movement of the trunk or crossing of the legs.

Record of Data

The average test was for a period of approximately 50 min. Consideration was given to a longer test period, but other demands upon the time of the subjects made it improbable that a sufficient number of volunteers could be obtained for any interval longer than that of a regular University class period.

The possibility of serious acclimatization error due to the short test period was investigated by testing for a 2½-hour period three subjects; in every case the results from the longer tests corresponded exactly with data from 50-min tests of

the same subjects under the same temperature conditions. In many of the later tests the subject's sensitivity to changing conditions was investigated by varying the temperature during the last 10-min interval. To permit expressing data from all tests on the same basis, and for a period during which conditions remained unchanged, the subject's reactions for the first and last 10 min intervals were not used for analysis, but four readings from consecutive 10-min intervals in the center of each test were selected.

As soon as the occupant was in the reflection enclosure the operator started recording temperatures and continued to make a complete record of all temperatures throughout the system at 5-min intervals during the 50-min test period. All temperatures were obtained from No. 36 copper - constantan thermocouples connected through a control board to a precision potentiometer which could be read to 0.001 millivolts. The potentiometer was checked against a standard cell at intervals during each test. Additional checks on temperature readings outside of the enclosure were available from a series of resistance thermometers and triple, 24-hour, recording thermometers.

Control of the ambient temperature was manual by adjustment of a throttling valve in the dichlorodifluoromethane suction line just ahead of the compressor; accurate control of the air temperature to within ±0.5 F of the desired setting was realized without difficulty. To maintain uniformity of external film coefficients on the enclosure and hence to keep the air-to-air temperature drop from varying, the volume of air circulated through the ambient space was held constant for all tests.

The psychological effect of divorcing the occupants from any direct knowledge of the existing air temperature was investigated, but no correlation was obtained between the feeling of warmth and the occupant's knowledge of air temperature. The conclusion was reached that the psychological factor, if existent, is too small to be of practical significance. All data from all tests were therefore used, without reference to the subject's knowledge or ignorance of the unusually low temperature in the enclosure.

Experimental Results

Of the 60 subjects used in this investigation, fifty were tested at enclosure air temperatures between 55 and 65 F. Of the tests 25 were conducted in air at ±1 F from 60 F and the remaining 25 were distributed over the ±5 F range in such a way that the weighted average for the 50 tests remained at 60 F. Use of a weighted average assumes a linear variation of the feeling of warmth with temperature; within the range considered and noting that 50 per cent of all tests were very close to the average temperature it is probable that this assumption is accurate within the limits of experimental error. The temperature of the enclosure inside surface was, in all tests, 3-4 F below the air temperature.

Results of the 50 tests based on 4 consecutive votes from the report of each different subject are shown in Table 1.

Table 1—Results of Tests Based on 4 Consecutive Votes from Report of Each Different Subject[a]

Schedule No.	Sensation	Votes Out of 200	Per Cent of Total
1	Uncomfortably hot	0	0.0
2	Uncomfortably warm	1	0.5
3	Comfortably warm	4	2.00
4	Ideally comfortable	94	47.0
5	Comfortably cool	83	41.5
6	Uncomfortably cool	18	9.0
7	Uncomfortably cold	0	0.0
		200	100.0

[a]All votes for reactions Nos. 2, 4, or 5 represent comfort conditions, and 90.5 per cent of all votes were for comfort.

The large block (41.5 per cent) of *comfortably cool* votes indicates that respiration of air at the lower temperature causes the average person to be conscious of a pleasant coolness even though the exterior body heat balance is perfectly normal. Many of the subjects recorded the impression that the air in the enclosure seemed to them unusually *fresh* and invigorating; some of the subjects read or studied between readings and commented, on leaving, that the enclosure atmosphere did not induce the feeling of drowsiness which they frequently associated with such activity. In contrast to the 22 subjects who commented favorably on the effect of breathing enclosure air, two men reported that the air seemed *stuffy*.

Analysis of Reactions by Subjects

The average reaction of each subject was determined from his four votes; *86 per cent of all subjects had an average which represented conditions of comfort.* The group average for 50 subjects, one test each, was 4.57; that is, half way between the reaction scheduled as ideally comfortable and that scheduled as comfortably cool. Investigating the distribution of votes cast by each individual, the variation of reaction during the test period is found to be as follows:

Of 50 subjects,

76 per cent were comfortable throughout the entire test period.

12 per cent were comfortable for 75 per cent of the time and uncomfortably cool or warm for the remaining quarter of the test.

12 per cent were comfortable for 50 per cent of the time and uncomfortably cool for the remaining 50 per cent.

0 per cent were uncomfortably cold or hot at all times.

100 per cent

Discussion of Results

A comment repeatedly occurring on subjects' data sheets is that an immediate sensation of warmth was experienced when the door of the reflection chamber closed. Many subjects noted a feeling of warmth on all exposed skin surfaces while some stated that this effect was limited to the face. A few tests were made with subjects having rolled sleeves; they reported that the exposed forearm remained comfortably warm for the duration of the test.

The most common report of local cooling was with respect to the ankles and feet of the male subjects. This was attributed to a draft near the floor of the enclosure, but this could not be detected with instruments. Tests of 4 female subjects disclosed an opposite reaction and suggested the possibility that the trouser leg may largely shield the ankle from radiation while exposing it to the cool air. Additional tests

to investigate this hypothesis are in progress.

Data from other Tests

Investigations with temperature conditions above or below those for comfort were not included in the objectives of the present work, which was concerned with comfort area. However, a few subjects were tested with enclosure air temperatures in excess of 65 F. The reactions they recorded, at temperatures above 66 F, included decided feelings of *stuffiness,* and, at 68 F, activity of the sweat glands became noticeable in two cases.

Six subjects were tested in air at 57 F with all surfaces at 52 F and with a supplementary heat source of 100 w; all these subjects were comfortable at all times, but the number tested is too small to allow consideration of these results as representative. Further, these six tests were performed under conditions which did not permit determination of the per cent radiant transfer from the 100-w source. Since the convective fraction from such a source would not in any way influence the occupant's feeling of warmth (air temperature was controlled by the operator and the convective fraction represents added cooling load), an analytical heat balance for this case is not possible.

Conclusions

Inside surfaces of low emissivity (high reflectivity) can be effectively used to reduce the heating requirements of an occupied room. This reduction is independent of the type of heating system.

Effectiveness of reflective walls varies inversely with room size and decreases rapidly with a reduction in the reflectivity of the surface. The equivalent reflectivity of a room may be low even though a large part of the inside surface is covered with highly reflective material; small *black body* areas, as open windows, door or grilles, severely reduce the equivalent reflectivity of the enclosure.

If polished surface insulation, as metal foil, is to be used in a given structure, and if architectural or other considerations will permit, an additional advantage will accrue from using the material on the inside surface rather than in the walls. When used in the interior of walls, reflecting surfaces reduce transmission losses; when used as an inside surface finish, the same materials not only reduce transmission losses, but act also to prevent net heat loss from the body and thereby permit realizing comfort with a lower air temperature and a reduced heating load.

Experimental data from 50 tests with 50 subjects established the possibility of attaining comfort in a reflecting enclosure with air at 60 F and surfaces at 57 F and with no heating of any kind other than that due to the occupants.

PEEBLES A PARTNER IN BASKERVILL

John K. Peebles, Jr., architectural engineer, has been admitted to partnership in the firm of Baskervill & Son, with offices in the Central National Bank Bldg., Richmond, Va. Mr. Peebles, a graduate of the University of Virginia, joined the Society in 1924 and for a number of years was a member of the firm of Peebles and Ferguson, Norfolk, Va.

F. M. YOUNG HONORED

F. M. Young, president and founder of Young Radiator Co., Racine, Wis., was recently elected an honorary member of Alpha Chapter of Pi Tau Sigma, honorary Mechanical Fraternity, at a formal initiation ceremony and banquet held at Memorial Union Building, University of Wisconsin, Madison, Wis. Mr. Young was honored by the fraternity in recognition of having *done much to advance the profession and to encourage young men to do likewise.*

NEW OFFICERS ELECTED

A. C. Buensod, president of Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., announces that M. S. Smith has been elected vice-president and will carry on his duties as General Manager, also that W. J. McDonald was elected Treasurer.

Mr. Buensod also takes pleasure in announcing that his associate, A. E. Stacey, Jr., is now Commander Stacey of the United States Navy and is on active duty somewhere in the Pacific. Mr. Stacey has resigned as officer and director of this organization for the emergency period.

WALLACE CONTINUES AS INDEPENDENT CONSULTANT

William M. Wallace, II, announces that he will practice as a consulting engineer for mechanical, electrical and sanitary work, with offices at 111 N. Corcoran St., Durham, N. C. This terminates his partnership association with Syska and Hennessy. Mr. Wallace received his B.S. in engineering from the University of Kentucky and his M.E. from the University of Kentucky Graduate School. He has been a member of the ASHVE since 1928 and is at present serving the North Carolina Chapter of the Society as its president.

HARRINGTON MANAGER OF MOTOR SALES

Elliott Harrington, sales manager of the General Electric air conditioning and commercial refrigeration department at Bloomfield, N. J., has relinquished the latter duties to become manager of sales of the Schenectady induction motor section of the G-E industrial department.

He has been connected with G-E air conditioning and automatic heating activities from their inception about 10 years ago and prior to that time had been in the industrial division from 1919. His early assignments on the development of an automatic oil furnace led directly to the formation of the air conditioning and commercial refrigeration department, in which he served in various engineering and sales capacities.

A native of Delavan, Wis., Mr. Harrington was graduated from Beloit College in 1916, and did postgraduate work in engineering at Massachusetts Institute of Technology and in business administration at Harvard. During World War No. 1, Mr. Harrington entered the U. S. Naval Air Service, serving 18 months as a pilot and on the development of an aerial bombsight, and acting as officer in charge of aerial gunnery and bombing at North Island air station in San Diego. He joined the ASHVE in 1930 and has taken an active part in its activities and meetings.

Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method

By Victor Paschkis*
New York, N. Y.

SUMMARY—By a new electric analogy method, the heat flow through two building walls was investigated. In one case with a 2-in. pine wall the temperature-time curve on the outside surface was known, while for the other, a 13-in. brick wall, the amount of heat radiated from the sun and the temperature of the surrounding air were known. In both cases, the heat flow from the inside surface to the air in the room, held at a constant temperature, was desired. The results obtained checked closely with the thermal measurements conducted at the ASHVE Research Laboratory

Introduction and Scope

IN order to choose the most suitable wall from a thermal consideration and to select either the cooling or heating equipment to maintain a prescribed inside air temperature for a given wall, it is necessary to determine the time-heat flow and the time-temperature curves.

The flow of heat through building walls has been the subject of many investigations conducted either by mathematical analysis or by direct experimentation.[1]

This report describes two experiments conducted by the *electric analogy method*[2]. Both experiments cover conditions of test previously conducted in the ASHVE Research Laboratory located in Pittsburgh, Pa. by a direct thermal method.

The analogy or model method of investigating transient heat flow problems is based on the identity of the equations governing transient heat flow and the flow of electricity in a resistance-capacity circuit. There is a perfect analogy between heat and electric phenomena as compiled in the following scheme:

Temperature .....Electric voltage
Heat flow (per unit time).....
.....Electric current

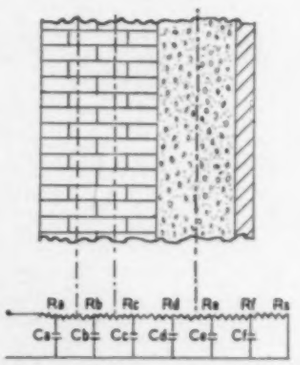

Fig. 1—Diagram of wall consisting of brick, concrete and plaster

The first part (*brick*) is considered to be cut into three sections; the second (*concrete*) in two; while the third layer (*plaster*) is represented by one section only. Each section is represented in the electrical model (wiring diagram under the sketch of the wall) by one resistor and one condenser. The resistors are marked R_a.....R_f; the condensers are marked C_a.....C_f. An additional resistor R_s represents the film conductance on one side of the wall. The other side of the wall is subjected to a given heat flow.

Thermal capacity (specific heat and density) .....Electric capacity
Thermal conductivity....Electric conductivity (reciprocal of electric resistivity)

It is therefore possible to arrange an electric circuit having certain characteristics, and to compute the desired heat values from electrical measurements. A non-mathematical description of this method has previously been published[3] and it will suffice in this discussion to describe the method by one example.

Assume a wall (Fig. 1), subjected to heat flow on one side, and the other side being in contact with air of constant temperature. In order to represent the wall in the electrical model (analogue), it is considered to be cut into sections, as indicated by the dotted lines in Fig. 1. Each section is represented in the model by an electric condenser (for the thermal capacity) and electric resistors for the thermal conductivity.

After arranging the model circuit, voltage is applied; if the temperature is constant, constant voltage is applied; if the temperature changes (*e.g.*, periodically) the voltage is also changed (periodically). There is a definite ratio between temperature and voltage, based on the ratio of electric and thermal resistivities and capacities, *e.g.*, 1 F may be represented by 2.5 volts. The current is measured and can be converted into thermal values expressed in terms of Btu per hour.

It is important to know that by appropriate choice of the electrical values, a *time ratio* can be selected, that is, a heat cycle of 24 hours can be *condensed* proportionally, to last in the model only 15 min. Or a very short heat cycle, lasting only fractions of a second, can be *stretched* to last several minutes.

Experiment A

The Problem

Given a building wall and the temperature-time curve for the outside surface. How does the heat flow from the inside surface change during one cycle, if the inside air temperature is held at a constant value?

The experiment was conducted for the conditions prevailing on

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*Research Associate, Department of Mechanical Engineering, Columbia University.

[1]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (ASHVE TRANSACTIONS, Vol. 38, 1932.)

Some Applications of Physics to Air-Conditioning Physics, by E. M. Pugh. (*Physics*, Vol. 7, March, 1936.)

Transient Heat Load in Calculating Air Conditioning Loads, by E. M. Pugh. (*Refrigerating Engineering*, Vol. 42, July, 1941.)

Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

The Flow of Heat Through Walls, by F. E. Giesecke. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

A Laboratory Method for Cyclic Heat Measurements on Walls and Roofs, by E. R. Queer and F. G. Hechler. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1941.)

[2]A Method for Determining Unsteady-State Heat Transfer by Means of an Electrical Analogy, by Victor Paschkis and H. D. Baker. (*ASME Transactions*, 1942.)

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

[3]Electrical Machine Solves Heat Transfer Problems, by Philip W. Swain. (*Power*, July, 1941.)

July 25, 1931 for a 2-in. pine panel described in a previous paper[4].

The physical data for this wall were as follows:

Actual thickness (including layer of plaster)—2.16 in.

Thermal conductivity—0.83 lb per square foot, per degree Fahrenheit per hour per inch.

Density—27.46 lb per cubic foot.

Specific heat—0.467 Btu per pound per degree Fahrenheit.

Inside film conductance—1.9 Btu per square foot per degree Fahrenheit per hour.

The thermal properties and the conductance were considered to be independent of temperature.

Procedure

The general procedure is described in the previously mentioned paper[5], but the present problem involved some special features which required some alterations in the equipment, explained later and referred to in Fig. 2.

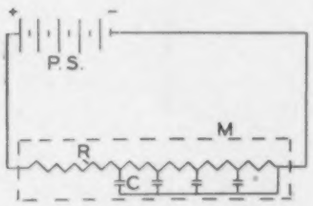

Fig. 2—Circuit diagram
P. S.=power supply
M=model
R=resistors
C=condensers

In a great number of transient heat flow problems, the heat flows in only one direction without reversing. Such heat flow problems can be represented by an electric model circuit, Fig. 2, where the potential of one end of the circuit is chosen as zero. The circuit itself consists of resistors and condensers which are respectively the analogues of the thermal resistance and the thermal capacity. For the present case the heat flow changes its direction during each cycle and the temperature-time curve swings periodically around a zero line, *e.g.*, the constant room (air) temperature.

The available recording voltmeters have the zero line at the left side of the scale, and thus do not allow the recording of voltage oscillating on both sides of zero. In order to use the instruments, the zero potential for the experiment had to be selected in such a manner that the whole oscillation of the cycle was on one side of the zero potential line. If Fig. 3 represents such a temperature-time cycle (or voltage-time cycle in the model), then the zero potential must be equal (or lower) than t_{min}. The circuit diagram of Fig. 4, conforms to this requirement.

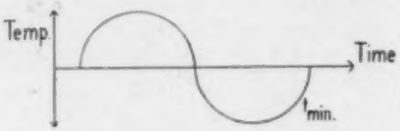

Fig. 3—Temperature-time wave

For the experiments with the model a time ratio was chosen. In the present case a ratio of 112.5 was selected; 1 sec in the model experiments represented 112.5 sec of heat flow through the wall. The daily cycle of 24-hour was thus represented by $\frac{24 \times 3600}{112.5} = 768$ sec.

The temperature-time cycle for the outside surface as drawn in Fig. 5 showed some irregularities, and these irregularities were imitated in carrying out the *model experiments*. The voltage representing the outside surface temperature was regulated according to the changes of the outside surface temperature, as recorded in 15-min intervals by the ASHVE Research Laboratory. Each 15-min interval was covered by two steps, representing 7.5 min each. The duration of each step in the experiment was $\frac{7.5 \times 60}{112.5} = 4$ sec, which is ample for setting the correct value.

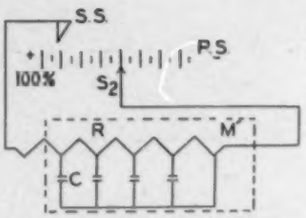

Fig. 4 — Circuit diagram for Experiment A
P. S.=power supply
S. S.=selector switch, taking any voltage from +(100%) to zero
S_2=switch taking voltage for constant room air temperature
M=model, consisting of *R* resistors, and *C* condensers.

The voltage-time cycle, representing the temperature-time cycle, was repeated seven times in order to make sure that two consecutive cycles were repetitions without any charging or discharging phenomena.

By means of recording voltmeters, the temperatures on the outside surface and on the inside

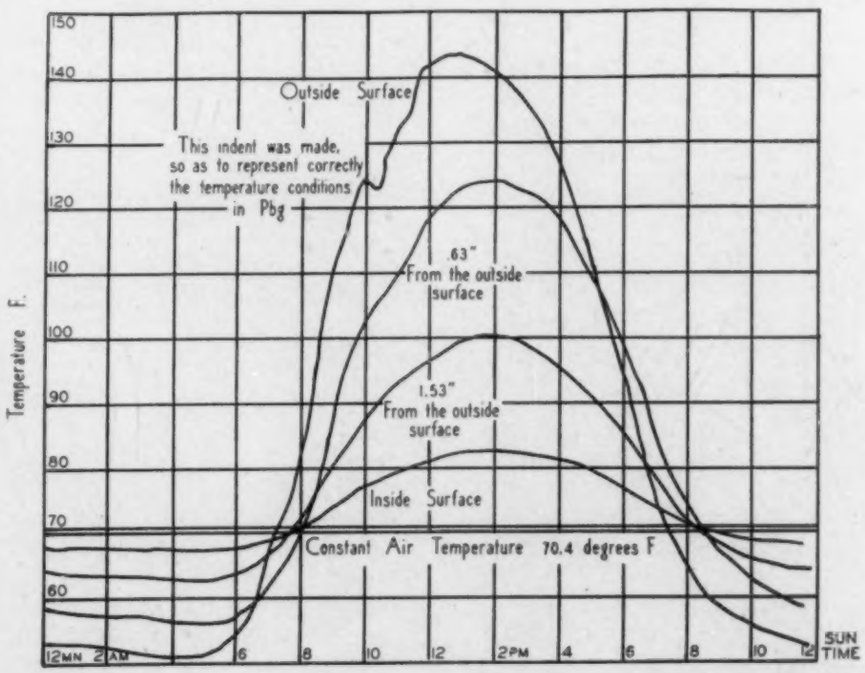

Fig. 5—Temperature-time curves; temperatures at the outside and inside surfaces of the wall and within the wall for Experiment A, 2-in. pine wall

[4]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (ASHVE Transactions, Vol. 38, 1932.)

[5]Loc. Cit. See Note 2.

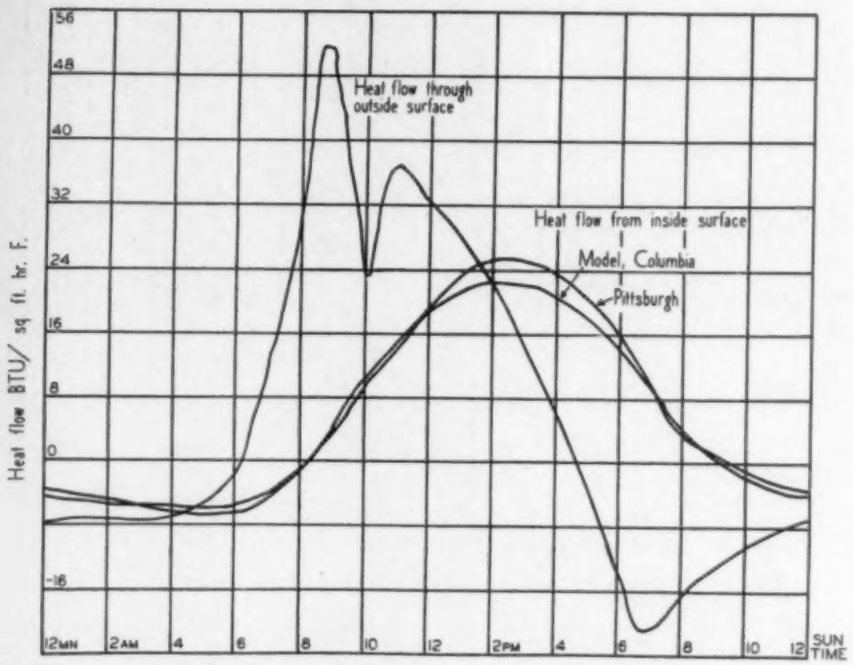

Fig. 6—Heat flow through outside surface and from inside surface; air temperature in room held constant at 70.4 F for Experiment A, 2-in. pine wall

surface were recorded. Furthermore, the temperature-time curves at points 0.63 in. and 1.53 in. from the outside surface were taken.

In intervals of 8 sec (corresponding to 15 min of heat flow) the current representing the heat flow entering the *outside surface* and the current representing the heat flow leaving the *inside surface* were read.

Results

The temperature-time curves are given in Fig. 5 with the time plotted as abscissa, and the temperatures as ordinates. Four curves were drawn: one for the outside surface; the next for a layer 0.63 in. from the outside surface; the next for a layer 1.53 in. from the outside surface; and the last for the inside surface. This chart illustrates clearly the decreasing amplitude and the lag of the temperature wave. The wave on the outside has the largest amplitude, occurring earliest, and the wave on the inside has the smallest amplitude, occurring latest.

The temperatures as shown were those obtained from the recording voltmeter. The curve for the inside surface was checked by measuring the current (representing the heat flow) from the inside surface to the air. These values, multiplied by the resistance representing the boundary (film conductance) equal the voltage difference between the *inside surface* and the *room air*. Hence, by adding to the product the (constant) value of the room-air temperature, the temperature of the inside surface may be obtained.

The curves in Fig. 6 give heat flow in relation to time. The time is plotted as abscissa, and the heat flow values as ordinates. The chart contains three curves: one for the heat flow through the outside surface as observed on the model; one for the heat flow from the inside surface (to the air in the room) as observed on the model; and a third for the heat flow from the inside surface as observed by direct thermal measurements at the ASHVE Research Laboratory.

The chart shows clearly the results of the mass effect of the wall. The crest of the heat flow as entering the outside surface is delayed and decreased in amount. The crest reaches the inside later and with a smaller amplitude.

The room temperature was held constant at 70.4 F. During the 24-hour period there was more heat flowing from the outside to the inside than vice versa. This *surplus* of *heat* was carried away by the cooling equipment which held the room temperature at 70.4 F. Without the cooling equipment in operation, the average temperature in the room would have been higher. Obviously, the heat flow (integrated over 24 hours) through the outside surface must equal the heat flow through the inside surface. Both values were measured independently; therefore a comparison of the two serves to check the accuracy of the method.

The total excess of heat flowing from the outside to the inside greater than the flow from the inside to the outside, as measured on the outside surface, was 123 Btu per square foot. Measurement on the inside surface showed a total heat of 117.5 Btu per square foot. The difference is 4.5 per cent.

Discussion of Results

1. The heat flow caused by the difference in temperature between the inside surface of the wall and the room-air (Fig. 6) shows greater deviations from the Pittsburgh results than the temperatures. This is due to the fact that the heat flow is proportional to the temperature *difference*, which is found by deducting from the changing surface temperature the constant value of the air temperature.

2. The difference between the results obtained by the model method and those obtained at Pittsburgh are to be attributed to the following causes:

a. In Pittsburgh the room temperature

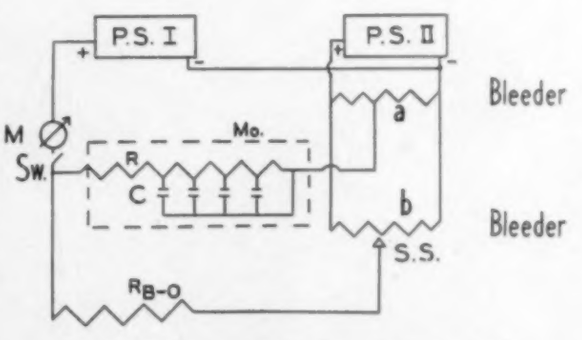

Fig. 7—Circuit diagram for Experiment B

P. S. I=power supply I, supplying current, representing heat flow from the sun; heat flow is varied from 100 per cent to zero
P. S. II=power supply II feeding the two bleeders (potentiometer) resistors, supplying the auxiliary potentials.
a=voltage corresponding to the constant room air temperature of 78.8 F
S. S.=selector switch to adjust voltage *b*
b=voltage corresponding to the outside temperature
M=milliammeter
Sw=switch
Mo=model
R_{B-o}=resistance representing the film conductance from the outside surface to the surrounding air

was not constant but was subject to variations of plus or minus 2 per cent. This small variation may have a big influence on the heat flow for the same reasons as explained in paragraph 1.

b. The film conductance is *not* constant but changes with temperature. The higher surface temperatures probably cause a higher film conductance and consequently higher heat losses.

c. Manual adaptation of the voltage on the outside surface might have caused a curve slightly different from the value measured in Pittsburgh.

Experiment B

Problem

Given a building wall and the amount of solar radiation in relation to time for the outside surface, and the temperature of the air surrounding the outside surface of the wall. How does the heat flow from the inside surface change during one cycle if the inside air temperature is held at a constant value?

The experiment was conducted for a 13 in. brick wall for which the ASHVE Research Laboratory gave the following physical properties:

Thickness—13 in.

Thermal conductivity—5.2 Btu per square foot per degree Fahrenheit hour, per inch.

Density—110.1 lb per cubic foot

Specific heat—0.196 Btu per pound per degree Fahrenheit.

Film conductance (outside)—4.0 Btu square foot per degree Fahrenheit per hour.

Film conductance (inside)—1.6 Btu per square foot per degree Fahrenheit per hour.

The thermal properties and film or air boundary conductance were assumed to be independent of temperature.

The outside film conductance of 4 Btu per square foot per hour per degree Fahrenheit occurs at a wind velocity of 6 mph (brick wall). The film conductance of 2 Btu per square foot per hour per degree Fahrenheit occurs at a wind velocity of 0.5 mph.

Procedure

The procedure was essentially the same as that used in Experiment A, the main differences being the following:

a. In order to represent the changing amount of heat input due to sun radiation, the current (milliammeter *M*, wiring diagram Fig. 7), was controlled.

b. The resistor R_{B-o} was used to represent the film or boundary conductance at the outside wall.

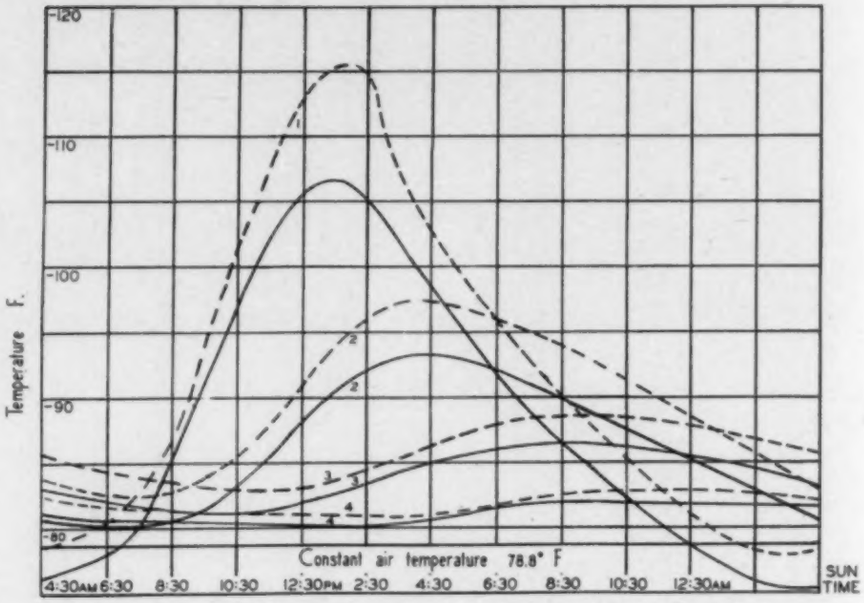

Fig. 8—Temperature-time curves; temperature at the outside and inside surfaces of the wall and within the wall for Experiment B, 13-in. brick wall

Full lines for an outside film conductance of 4 Btu per square foot per hour per degree Fahrenheit. Dotted lines for an outside film conductance of 2 Btu per square foot per hour per degree Fahrenheit

Lines:
1. at the outside surface
2. 4.33 in. from outside surface
3. 8.67 in. from outside surface
4. at the inside surface

c. One end of the resistor R_{B-o} was connected to the end of the model representing the outside surface, the other to the power supply (*P.S.* II), supplying a changing voltage. This voltage was regulated according to a curve representing the outside air temperature.

d. As long as energy was being received from the sun, it was split into two parts: one flowing into the wall, and the other from the wall surface to the surroundings. After the radiation from the sun ceased, switch *Sw* was opened; and the wall was only in heat exchange with the surrounding air by means of the resistor R_{B-o}.

In order to get some indication as to the importance of the outside film conductance, a second set of experiments was run, with a boundary resistance representing a film conductance of only 2 Btu per square foot per degree Fahrenheit per hour.

Results

The results are represented in a manner similar to those in Experiment A. The temperature time curves are shown in Fig. 8 with the time plotted as abscissa, and the temperatures as ordinates. Two groups of curves were drawn. One group, drawn with full lines, shows the temperatures for an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour; the other, drawn with dotted lines, shows the temperatures for an outside film conductance of 2 Btu per square foot per degree Fahrenheit per hour. Each group consists of four curves: one for the outside surface; the next for a layer 4.33 in. from the outside surface; the next for a layer 8.67 in. from the outside surface; the last for the inside surface.

The lower portion of Fig. 9 shows the heat flow from the inside surface to the air in the room in relation to time. In the upper part of Fig. 9 the temperatures are plotted in relation to time. Time is plotted as abscissa for both sets of curves, while heat flow and temperatures are plotted as ordinates.

Three heat flow curves were drawn for the following conditions: one obtained on the model, for 2 Btu per square foot per degree Fahrenheit per hour outside film conductance and constant room air temperature of 78.8 F; one for the same previous conditions, but for an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour; and one obtained in Pittsburgh on July 21, 1940, with an average outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour, with the air temperature of the room being controlled by a thermostat, and

oscillating as shown in the curves in the upper part of the chart.

Two temperature curves are shown in Fig. 9 with the temperature scale starting at 77 F. One indicates the inside surface temperatures as measured on the model, for an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour. This curve is an enlargement of the corresponding one shown in Fig. 8. The other curve shows the oscillations of the air-temperature in the room during the Pittsburgh tests on July 21, 1940.

Discussion of the Results

1. The amplitude of the temperature curve on the outside surface is flattened considerably by the thick wall so that the amplitude on the inside is much smaller. With an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour, the amplitude of 106.6 — 75.1 = 31.5 F on the outside is flattened out to 81.9 — 80.1 = 1.8 F on the inside, or 5.7 per cent of the outside surface amplitude. With an outside film conductance of 2 Btu per square foot per degree Fahrenheit per hour, the amplitude of 115.5 — 78.3 = 37.2 F on the outside is flattened out to 82.5 — 80.8 = 1.7 F on the inside, or 4.5 per cent of the outside surface amplitude.

2. The outside film conductance is of great influence on the heat flow from the inside surface, if the outside surface receives its heat as sun load. The total heat flow from the inside surface to the air in the room, which is kept at a constant value of 78.8 F by means of the cooling equipment is:

with a film conductance of 2 Btu per hour per square foot per degree Fahrenheit—437.2 Btu.

with a film conductance of 4 Btu per hour per square foot per degree Fahrenheit—312.9 Btu

The ratio $\frac{437.2}{312.9} = 1.4$; that means an increase in outside film conductance from 2 to 4 (corresponding to an increase of wind velocity from 2.5 to 6 mph) reduces the cooling load to 60 per cent of its original value (at 2 Btu per hour per square foot per degree Fahrenheit). The reduction is not the same over the entire cycle of 24 hours. The discrepancy is largest at 1:15 p.m. (time of smallest heat flow): $\frac{3.02}{1.87} = 1.62$; the discrepancy is smallest at 10:45 p.m. (time of greatest heat flow) = $\frac{6.01}{4.61} = 1.30$. These experiments bring out clearly that the heat flow is subject to a much higher percentage variation over the 24 hours than the inside surface temperature.

3. The results of the model experiments can be compared in two aspects with the direct thermal measurements: a. The outside surface temperature, and b. The heat flow from the inside surface to the air in the room.

Outside Surface Temperature

Inasmuch as only the heat flow from the sun was given, the outside surface temperature results from the heat flow through the wall and to the outside air. The temperatures as measured in Pittsburgh and those found by the model method check closely. The largest discrepancy is 4 F, the average discrepancy is only 1.9 F.

Inside Heat Flow

The heat flow as observed in Pittsburgh does not check closely with the values measured by the model method. The minimum measured in Pittsburgh was 1.35 Btu per square foot per hour per degree Fahrenheit as against 1.87 as measured on the model. The values for the total amount of heat gain (heat flow integrated over a 24-hour period) differ only slightly. The *Pittsburgh value* is 290.98 Btu per 24 hours, while the *Columbia value* is 316.78 Btu per 24 hours.

The difference in shape of the two curves may be attributed to the difference of conditions prevailing during the two experiments. In Pittsburgh the room-air temperature was oscillating, while the model experiment was conducted with constant air temperature. Naturally the heat flow from the inside surface to the room-air is the result of the difference in temperature between the two values.

The shape of the temperature wave on the inside surface is almost independent of the small fluctuations of the room-air temperature and is caused entirely by the nature of the wall and the heat wave imposed on the outside surface. This temperature wave is in interaction with the wave of the air temperature. It is obvious that the resulting heat flow must differ from the flow obtained from the difference between the wave of inside surface temperature and a constant air temperature. This is still stressed by the fact that the temperature differences causing the heat flow are of the same order of magnitude as the oscillations in air temperature.

The differences in heat gain (heat

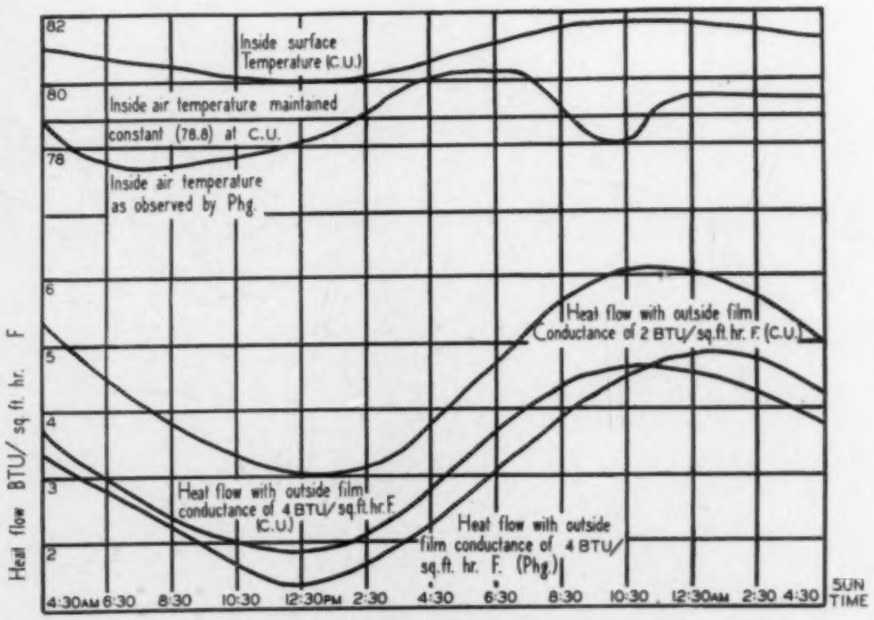
82
80
78
6
5
4
3
2
Inside surface Temperature (C.U.)
Inside air temperature maintained constant (78.8) at C.U.
Inside air temperature as observed by Phg.
Heat flow with outside film Conductance of 2 BTU/sq.ft.hr.F. (C.U.)
Heat flow with outside film conductance of 4 BTU/sq.ft.hr.F. (C.U.)
Heat flow with outside film conductance of 4 BTU/sq.ft. hr. F. (Phg.)
Heat flow BTU/ sq ft. hr. F
4:30AM 6:30 8:30 10:30 12:30PM 2:30 4:30 6:30 8:30 10:30 12:30AM 2:30 4:30
SUN TIME

Fig. 9—Results from Experiment B, 13-in. brick wall

Upper curves; inside air temperature
Lower curves; heat flow from inside surface to air in room

flow integrated over a 24-hour period) can be explained by several factors. First, the inside film conductance assumed to be 1.6 Btu per square foot per hour per degree Fahrenheit might have actually been another value. The heat flow being directly proportional to the film conductance, would have had correspondingly different values. Furthermore, the measuring of small temperature difference (only 1 to 3 F) is subject to larger proportional errors than the measurement of larger differences. It is believed that this error is greater in case of the heat flow meter than in case of the model measurement.

Conclusions

The model method yields results which check closely with direct thermal measurements. Advantages of the model method are:

1. Elimination of building actual walls.
2. Considerably shorter time for experiments.
3. Elimination of the dependence on weather.
4. Possibility of maintaining constant air temperatures.
5. Possibility of investigating separately the influence of the different variables (*e.g.*, film conductance, etc.)

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Campbell, Alfred Q. Jr., Lieut., Field Artillery.
Campbell, George W., First Lieut., Army Air Corps.
Caskey, Luther H. Jr., Second Lieut., Personnel Officer.
Cheeseman, Evans W., Lieut., Personnel Adjutant.
Clemens, Joseph D., 2nd Lieut. (Engrs.) Air Corps.
Croley, Jack G., First Lieut., 76th Coast Artillery.
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Dickson, Robert W. Jr., First Lieut., Infantry.
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*Compiled from information supplied for Roll of Membership as of January 1, 1942.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Physiological Fatigue from Exposure to Hot Environments

By M. B. Ferderber, M.D.,* F. C. Houghten,** Pittsburgh, Pa., and Walter L. Fleisher,† New York, N. Y.

SUMMARY—Severe and continued exposure to heat has been shown by studies of fever therapy and industrial conditions to affect the temperature—white cell relationship of the blood. This is shown to be fatigue of physiological response, which is not only experienced in these applications but in many hot industrial and climatic conditions.

A COMMUNITY of interest has developed during the past two decades between the application of heat by the medical profession in the treatment of disease and the studies of thermal relations between man and his atmospheric environment, as carried out by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The early studies[1,2,3,4] of physiological reactions of persons to hot atmospheres by the ASHVE, in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service, have been accepted by the medical as well as the engineering profession. The later studies of thermal exchanges[5,6,7,8] between the human body and its atmospheric environment as affected by the temperature, moisture content and movement of the air, and the application of air conditioning to hot industrial environments[9,10,11] by the Society were all of some little value as a background in the application of saturated conditioned air to fever therapy.[12, 13]

The past several years have seen extensive developments in fever therapy and other heat treatments of disease by the medical profession which has greatly extended the knowledge of the effect of hot atmospheres, either in industry or in tropical climates, on the physiological response, comfort and well-being of man.

This paper deals with some of the clinical findings resulting from fever therapy, with an attempt to interpret their significance to the reactions of normal persons subjected to long exposure to heat, either in industry or in the tropics. Important findings in these studies have been the numerical increase in the white cell or leucocyte count of the blood following the application of fever therapy,[12,13] the relation of this increase to the clinical response, and

*Research Fellow, Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, and Advisor on Physiological Studies at the ASHVE Research Laboratory. MEMBER of ASHVE.

**Director of ASHVE Research Laboratory, and Associate Fellow, Department of Industrial Hygiene, School of Medicine, University of Pittsburgh. MEMBER of ASHVE.

†Chairman, ASHVE Technical Advisory Committee on Air Conditioning in Industry. MEMBER of ASHVE.

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[12]Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937.)

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the observed fatigue, resulting in a failure in this increase in certain cases. This experience accounts for the unsatisfactory response of normal persons exposed for long periods to high temperatures, either in hot industries or hot tropical climates, and has a very important industrial hygiene significance. A better understanding of these cases and the advantage of improving them should be obtained through the application of air conditioning.

From an analysis of 847 patients treated in a cooperative study it was possible to verify certain conclusions regarding the relationship between white count response, temperature elevation, and improvement. The vague reference to *resistance, body immunity, tolerance,* etc., is not entirely clear even to the writers, but when lack of improvement fails to occur in fever treated patients, these repeated experiences have been very helpful in determining whether treatment should be continued or discontinued.

Clinically, the role of the leucocyte as it appears in so-called normal states of health from an accepted text book would be approximately 6,000 with an average of 60-70 per cent polymorphonuclear leucocytes, 25-33 per cent lymphocytes, 2-6 per cent monocytes, 1-4 per cent eosinophiles, 0.25-0.50 per cent basophiles. Disease, fever therapy, or exposure of humans to hot environments either by occupational or coincidental adaptation, may alter these figures over a wide range, and such alterations may apply to the temperature variations.

Except for such diseases as malaria, typhoid, measles, influenza, etc., it is most common to find an increase in the total white count accompanied by a rise in body temperature. This phenomenon is frequently described as *resistance to infection* or the *protective power of the human organism.* An excellent example of this action was observed in pneumonia research, where those patients with leucocytosis (increased leucocytes) and high temperatures withstood the disease more satisfactorily than those having leucopenia (reduction in leucocytes) or no change along with a slight rise in body temperature. If this assumption is correct, it may be possible to expect a greater average of improvement with high white counts and fever than in any other group. This is illustrated in Fig. 1, which shows the index of improvement in proportion to the cellular increase before and after treatment.

Those diseases which have responded best to fever therapy are gonorrhea, syphilis, chorea, and certain forms of arthritis. However, under certain conditions where dis-

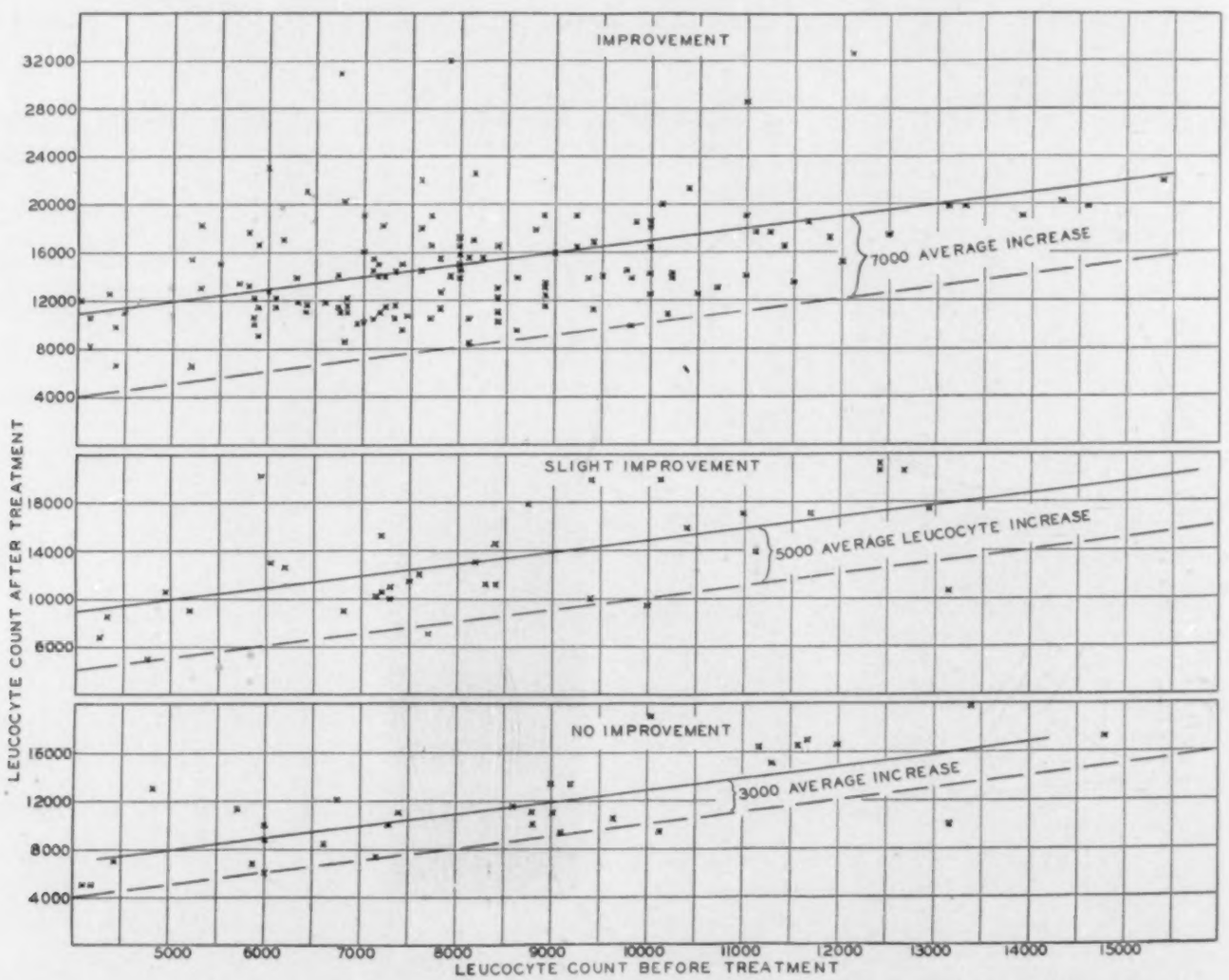


Fig. 1—Relation between degree of improvement, the normal leucocyte and increase in leucocyte count during 3-hour treatment in about 108 F saturated air

ease is not a factor, certain definite changes have been observed in relationships between hot environments and the temperature-white cell ratio, which will be discussed later in this report. Using identical criteria for all patients treated, the following may be postulated:

1. Failure of white cell increase indicated lack of clinical improvement.
2. Difficulty in body temperature elevation produced the same outcome.
3. The resistance to heat revealed failure in improvement in addition to lack of temperature and white cell increases.

The white cell increase was arbitrarily set at a minimum of 50 per cent of the pre-treatment figure as some indication of improvement, although a greater increase is more satisfactory. The optimum temperatures necessary for clinical improvement should fall within standardized ranges of 106-107.4 F for gonorrhea; 104-105 F for syphilis; 105 F for chorea, and 103-105 F for nonspecific arthritis.

Fever therapy has proved very helpful in gonnococcal infections, especially in its earlier stages. In long standing cases, temporary relief resulted where failure of leucocytic and temperature increase occurred, either after the first or subsequent treatments. Even in early cases, where fever was administered frequently at short intervals patients have become *resistant* or *immune* to heat, in addition to little or no change in white count.

During the first World War, Wagner-Yauregg inoculated malarial organisms into the blood of patients suffering from syphilis to produce fever. More recently, fever therapy has to a great extent eliminated this method for reasons of safety, control and flexibility. Here, too, the ratio of improvement to increases in temperature and white cell count previously described holds true in every respect.

Chorea, especially in children, responds quite satisfactorily to fever therapy. They too, reveal the same tendencies of white blood count, temperature relationship as the adult.

However, the arthritic presents a very interesting situation. The causes in many cases are vague, except where known infection can be demonstrated. Since controversy exists over this disease only one group shall be discussed; i.e, infectious arthritis. Usually, an early case is characterized by fever, increased white cell count, swollen, painful joints and prostration. Frequently, fever is beneficial in the majority of patients, and even though no specific measures are taken the described findings usually subside. Later, there may be a recurrence of the same illness, with smaller increases in temperature and white cell count, and treatment becomes even more difficult. These flare-ups may occur occasionally, and it has been observed that hyperpyrexia is of little value once it has become chronic. In this stage, one finds the tender, enlarged joints without increases in fever or white count, and fever therapy apparently is of little or no value. From this, it would appear that long continued illness, however slight, would exhaust those necessary powers required for eliminating such infections. Treatment, therefore, can be directed toward the early illness before a deficiency of physiological factors occurs.

Discussion

The experiences in these studies and those of many other investigators definitely indicate the physiological changes produced by exposures to hot environments. Since such changes may be accepted as a definite part of fever therapy, it would follow that exposures, experimental, occupational or industrial, of humans to hot atmospheres should produce some physiological changes. In 1938 the Society, with the help of the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, investigated[14] the physiological effects of hot atmospheres on human beings. These experiments were based upon the exposure of young healthy adults to environments which might be found in occupations existing in those sections of the country where high dry-bulb temperatures occur. It was observed that working in high effective temperatures of approximately 85 to 90 deg would produce increases in pulse rate, body temperature, and finally the leucocyte count. This latter finding was based on the original fever therapy work conducted[15] at the Society's laboratory, when individuals were placed in saturated psychrometric rooms. One experience is interesting, since it concerns a young Italian male subject, age 23, who showed neither temperature nor white cell increase and because of his extreme restlessness was removed from the hot chamber. He did not admit to the presence of any chronic infection until questioned, although his blood count and temperature were perfectly normal before the experiment began. Since this subject resembled patients with chronic infections previously described in this report, the first laboratory experience with individual resistance to heat was observed.

There is a normal diurnal variation in body temperature, which is usually greater in sick people; and, therefore, since the increase in white count is a function of increased body temperature, this subject must have had daily leucocytic rises. However, should this daily rise and fall continue for a very long period, it is not unreasonable to assume that a state of exhaustion or fatigue of the defense mechanism of the body would occur. This may be compared with the elementary physiology of stimulating frog muscles to a state of fatigue where no further action occurs due to overstimulation. If one grants the logic of this reasoning, it is not unlikely that long exposures to high environmental temperatures might not be conducive to good defense against infection. This may be better illustrated by experiences with pneumonia patients who have been infected during the hot summer months. In the cold months the same procedures are used, including serum, cooled oxygen tents, chemical treatment, etc.; and yet the so-called *summer* pneumonias are feared for some reason that is yet intangible. The troublesome *summer* cold is another example of failure to respond during long continued heat exposures. The same analogy may be used in any other infection; for example, gonorrhea or arthritis, where there was probably a daily rise in fever and white count, but little or no response to fever therapy previously described. On the other hand, the early case may respond satisfactorily

[14] Loc. Cit. Note 10.

[15] Loc. Cit. Note 12.

but repeated exposures to hot atmospheres failed to produce an increase in temperature and cell count. In this case the conditions of exposure have artificially produced a status of fatigue or exhaustion of the blood cell producing (hemopoetic) and thermal regulating mechanism of the body.

From the laboratory standpoint, including the experimental industrial work, the question might be raised whether this cell count increase was due entirely to dehydration which in turn produced blood concentration. Both capillary and venous blood counts were made; and since there was little or no difference, capillary blood samples were continued to be used. The work of Krusen[16] amply illustrates the changes which do occur in the count before and after fever therapy. If dehydration is prevented by sufficient fluid intake during the treatment, there is little or no variation in hemoglobin or cell content of the blood. Differential counts on the white cells also verified the absence of dehydration. There is a relative and absolute increase of polymorphonuclear leucocytes either during or immediately after the febrile period. This numerical increase has been as high as 30,000 from a pretreatment figure as low as 6,500.

When one considers the role of the leucocyte and its use in infection, it becomes clear that its phagocytic power (ability to destroy micro-organisms or harmful cells) is all important, either by destruction of the offending organism or removal of the products of infection. The original work of Metchnikoff[17] capably demonstrated that the polymorphonuclear leucocytes take up the bacteria into the cell, demonstrated by staining, which shows vacuolization where the organism formerly existed. Metchnikoff has also shown that phagocytosis is regularly more active in cases where animals or humans eventually recover. Such a statement would tend to bear out the investigations here reported that there is a definite relationship between the extent of improvement and phagocytosis interpreted as white cell increase.

[16]The Blood Picture Before and After Fever Therapy by Physical Means, by F. H. Krusen. (*American Journal Medical Science* 193:470, April, 1937.)

[17]Zinsser, Bayne-Jones Text Book of Bacteriology, 232-234, 1934, by E. L. Metchnikoff. (D. Appleton-Century Co.)

Fever therapy and chronic infection are not the only producers of static cell and temperature responses, but high temperatures may produce the same conditions. Although individuals have an excellent thermal regulatory system, exposure to hot environments by occupation or residence produces elevated body temperatures and in turn increased white cell counts. Again, this situation is not unlike a chronic infection because the exposure is daily and similar to the diurnal temperature variation. Therefore, in cold weather, after working in hot spaces, removal to a cool environment permits a return to normal, but in very hot weather the change is either retarded or the place of residence is not sufficiently cool to produce a return to normal. The term *acclimatization* is fitting because the worker becomes accustomed to physiological changes where the working environment is not unlike the resting environment. It has been shown[18] that there is a difference in response in different seasons of the year. From workers who have spent long periods in very hot climates, there is evidence to show that the failure of physiological defense against infections is directly proportional to the time spent in the area. If true, one might conclude that the constant existence in extremely hot environments might produce a condition in the body where temperature and white cell response might fail as a defense measure.

It is frequently desirable to alleviate undesirable exposure to these hot atmospheres. Means for accomplishing this end have been established through research. Where the occupancy density in a hot industry is high, the most logical and economical is to air condition the entire enclosure so as to supply ambient air in the entire enclosure sufficiently satisfactory to give the desired results. In industrial establishments where a small number of workers are engaged in large enclosures complete air conditioning may not be feasible. Here, one of the following methods may be used as indicated in earlier laboratory studies:[19] (1) In some instances the worker may be placed in a conditioned booth from which he may observe operations and perform his required tasks. (2) The worker may be supplied with a blast of cool, high velocity air into which he may step, thus providing himself with the degree of cooling required. (3) Cooled air may be blown through a loose-fitting, relatively air-tight suit so as to cool the worker without affecting the surroundings.

[18]Loc. Cit. Note 11.

[19]Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet. (ASHVE JOURNAL, *Heating, Piping and Air Conditioning*, July 1941, p. 462.)

AWARD TO McDERMOTT

Paul F. McDermott, research engineer with the Johns-Manville Research Laboratories, Manville, N. J., was awarded the *ASRE* Wolverine Award for 1941. This award, which is presented annually by the *American Society of Refrigerating Engineers* for the best paper published in *Refrigerating Engineering*, was conferred upon Mr. McDermott for his paper entitled Moisture Migration—A Survey of Theory and Existing Knowledge, which he presented before the *ASRE* 28th Spring Meeting in Cincinnati, Ohio, on May 28, 1941, and which was published in the August issue.

ASRE ELECTS OFFICERS

At the 37th Annual Meeting of the *American Society of Refrigerating Engineers* held at the Jefferson Hotel, St. Louis, Mo., December 2-4, 1941, it was announced that the following were elected to serve as officers for the coming year: *President* — W. R. Hainsworth; *Vice-Presidents*—C. R. Logan and A. B. Stickney; *Treasurer*—John F. Stone; *Secretary*—David L. Fiske.

A record attendance was reported at this 3-day meeting which emphasized problems of the industry in connection with the defense program, and in sessions on government specifications and talks on cooperation with the OPM, it was realized that rapid adjustments must now be made in the refrigeration and air conditioning field.

One of the special features at the meeting was the *ASRE* Distinguished Service Award, the first in the history of the Society, which was presented to W. S. Shipley for his important work in national defense. Mr. Shipley is chairman of the board of York Ice Machinery Corp., York, Pa., and developed the York Plan.

Sanitary Ventilation†

By William Firth Wells,* Philadelphia, Pa.

SUMMARY — Recognition of the importance of air disinfection in ventilation is presented with a description of improved techniques for eliminating microorganisms by ultraviolet radiation. Data obtained during a contagious epidemic in three Philadelphia schools demonstrates the hygienic value of sanitary ventilation. Further studies of air purification in large enclosed atmospheres are emphasized to eliminate hygienic hazards from air supplies.

PRIOR to the streamlined air conditioned era, heating for comfort and ventilating for health were accepted notions. Ventilation of heated confined spaces, vitiated by occupancy, was regarded as a health essential. Pettenkofer, the father of hygiene, proposed in 1858 the familiar carbon dioxide method for measuring the atmospheric density of occupants. Dilution of exhaled carbon dioxide with 20 cu ft of air per person per minute yields approximately one part per thousand. DeChaumont (1875) perceived an unpleasant odor in confined spaces when exhaled carbon dioxide exceeded six parts per ten thousand, for many years accepted as a limit by sanitarians.

Noxious vapors and exhaled effluvia are no longer considered infectious. Though bacteriology and the germ theory of disease were born together with Pasteur's classic experiments on bacteria in habited atmospheres, perfection of bacteriologic techniques during the last two decades of the 19th century identified specific agents of disease, and disclosed in rapid succession their transmission by other routes than air. Failures to recover nasopharyngeal organisms on plates exposed beyond an arm's length of a coughing or sneezing person convinced Flügge that infected droplets settled out in short distances and in brief time intervals.

Sanitary bacteriology during the next 20 years demonstrated general bacterial contamination, but successful sanitary control of infections ingested in water, milk or food, or conveyed by insects, was not extended to respiratory infections. These *catching* diseases might perhaps spread without vehicles, for obviously direct transfer by personal contact was possible among aggregations within enclosed atmospheres.

Discarding Pettenkofer's teachings, the ventilation engineer pronounced fit to breathe air which satisfied comfort requirements, and turned to factors of humidity, temperature and air motion to explain the physiological effects of vitiated air. Environmental control techniques soon met these new specifications of ventilation, now called air conditioning.

New evidence gathered during the last 10 years, with new instruments and techniques, has reopened the case against air as a vehicle of infection. Better understanding of state of air-suspended infection, the role of droplet evaporation, and the behavior of viruses in respiratory contagion, dispelled many cherished fallacies. The effect of ventilating factors upon the dispersion and viability of organisms in passing from person to person in an enclosed atmosphere was studied by suspending test organisms at one point and sampling at another. Improved methods of disinfection eliminated microorganisms from semi-enclosed atmospheres at 10 to 100 times the rate practically attainable by mechanical means of ventilation. Since disinfection rate depends upon the type of organism and the state of suspension, definition by comparison with removal rate by air replacement strictly applies only to a particular organism. Elimination of a standard suspension of a standard organism, equivalent to the removal by one air change, defines the lethe.

The peculiar vulnerability of air-suspended microorganisms to photobiotic effect of ultraviolet radiation offered practical means to test the hypothesis that the semi-enclosed atmospheres of our habitations in cold weather provide the vehicle for the epidemic spread of contagion. During last winter the largest epidemic of measles in the history of Philadelphia provided unusually significant epidemiologic data on the protection afforded by radiant disinfection in three primary schools. An intensive study of sanitary ventilation in the schools was undertaken to determine the relationship between sanitary and hygienic performance, between dosage and prevention. Interpretation of radiation into sanitary ventilation defined by these studies is outlined in Table 1 and Fig. 1.

Total lethal flux (in milliwatts)

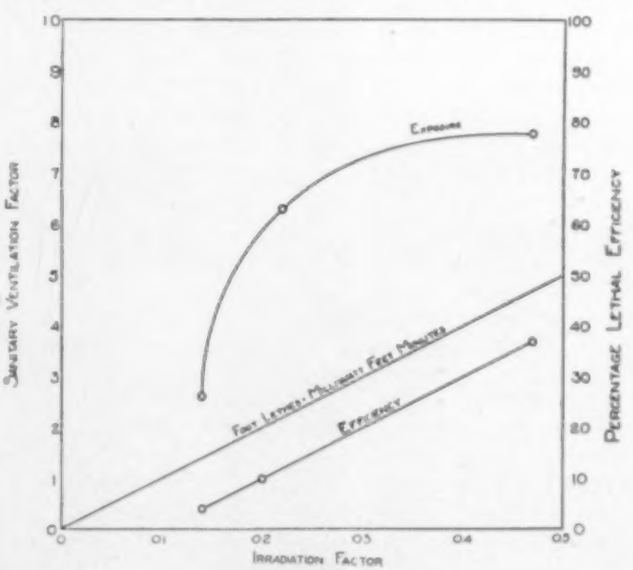


Fig. 1—Disinfection factors

Relation between exposure and irradiation factors, and relation between irradiation and sanitary ventilation factors. Exposure factor approaches one as irradiation and sanitary ventilation factors approach one-half because approximately one-half either of the radiant distance of unreflected radiation, or of the total lethal flux absorbed by reflectors, is unavailable for irradiation.

†From the Laboratories for the Study of Airborne Infection, supported by a grant from the Commonwealth Fund.

*Associate Professor of Research in Airborne Infection, Department of Public Health and Preventive Medicine, University of Pennsylvania School of Medicine. MEMBER of ASHVE.

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

Table 1—Sanitary Ventilation

Fixture	Germantown Central	Swarthmore Wall	
Source	Suspended Indirect	Beam Elevation 60 Deg	Beam Elevation 25 Deg
Volume (cu ft)	6,000	7,000	7,000
Total lethal flux (mw)	2,000	20,000	20,000
Uniform cubic flux density (mw/ft³)	6.06	54.7	54.7
Average flux density (mw/ft³)	1.26	7.64	25.5
Lethes per minute	0.8	2.0	20.0
Factors			
Irradiation factor	0.21	0.14	0.47
Exposure factor	0.63	0.26	0.78
Sanitary Ventilation factor	0.13	0.04	0.37
Sanitary Equivalent Pure Air Replacement (cfm/Occupants)			
Foot-lethes per occupant per minute	200	500	5,000

Table 2—Lethes Per Hour[a]
(Between Various Points)

Position	Swarthmore Bench Level	Swarthmore Upper Level	Swarthmore Reflector Horizontal	Germantown Bench Level	Germantown Upper Level
A-1-1	61.0	.....	.....	30.5	.....
A-1-2	91.7	.....	.....	35.9	.....
A-1-3	120.0	.....	.....	47.6	.....
A-1-4	.....	.....	.....	37.5	.....
B-2	32.8	1054	.....	.....	.....
B-1	52.9	293	635	36.7	80.0
B-3	75.0	.....	.....	.....	.....
B-4	42.9	.....	352	.....	.....

[a] Wells, W. F. and Green, R. Bacteriologic Study of Radiant Disinfection of Air in the Control of Epidemic Spread of Contagion. (Paper read before American Public Health Association, Oct. 14, 1941.)

represents lamp ratings given by the manufacturer. Uniform cubic flux density represents the maximum flux density (in milliwatts per square foot) obtainable in a cube of the given volume by the given flux. It is obtained by dividing the total lethal flux by the cube root of the volume squared. Generation of lethal flux is an electrical problem.

Average flux density is obtained by dividing milliwatt feet of actual irradiation by the volume. Radiant flux (in milliwatts) multiplied by distance (feet) through which that flux acts gives irradiation in milliwatt feet. The same lethal flux emerging from the same fixtures located in the same position in the same room, giving the same uniform cubic flux density, thus gives entirely different average flux density when intensity distance differs because changing angle changes the ray length intercepted by the enclosing surfaces. Ray length is thus as important in irradiation design as ray strength.

Lethes per minute are a sixtieth of equivalent air changes per hour. Radiant disinfection obviously is not determined solely by the quantity of non-uniformly distributed radiation. Thus, lethal change due to irradiation of recirculated air depends primarily upon rate of recirculation, which cannot be exceeded, regardless of the quantity of radiation applied in a duct. Circulation between more and less strongly irradiated zones in a room likewise determines lethes per minute. The lethal effect of a design must be determined bacteriologically in derivation of disinfection factors. Air motion and equivalent air change offer problems in ventilating engineering.

The irradiation factor is derived by dividing average flux density by uniform cubic flux density. Since the irradiation factor varies with radiation footage, it measures effectiveness of design. Conversion of radiation into irradiation is not the responsibility of the doctor or the lamp manufacturer, but of an irradiation engineer who can unite ventilating, illuminating, and sanitary engineering into one hygienic objective.

The exposure factor is derived by dividing lethes per minute by the average flux density. This also is found not to be directly proportional to the irradiation factor (Fig. 1), though dependent upon it in two ways. First, the more nearly irradiation becomes uniform through an enclosed space, the greater will become the exposure factor, regardless of air circulation. Uniformity of exposure of organisms circulating from larger, more irradiated, through smaller, less irradiated, portions of a room, is a second, more important advantage. The former is a problem in light distribution, but uniformity of exposure due to circulation is largely a problem in ventilating engineering.

The sanitary ventilation factor becomes the product of the irradiation and exposure factors. It approaches a half rather than one because approximately one-half, either of the radiant distance of unreflected radiation, or of the total lethal flux absorbed by reflectors is unavailable. Exposure of the test organism to a uniform intensity of a milliwatt per square foot would yield a lethe per minute if irradiation and sanitary ventilation factors are equal. Actual determinations are not yet accurate enough to establish the biophysical constant relating physical and bactericidal units, but the factor still offers a practical means of defining radiant disinfection. Sanitary specifications can be translated by this factor into irradiation.

Foot lethes per occupant per minute express equivalent sanitary ventilation. Organisms circulating between two points would be removed at the same rate as from a room by replacement with an equivalent number of cubic feet of pure air per occupant per minute. The order of magnitude of these values, however, is much higher than would be practicable by actual air change during cold weather. Values hitherto unattainable become practicable where rigid requirements demand and disciplinary measures permit protection of the occupants against overexposure to radiation. Interpretation of infection and disinfection in terms of air conditioning is a sanitary problem.

Lethes per hour between different points in a room, shown on Table 2, take on added significance in the light of these considerations. More uniform distribution of light at Germantown compensates in large measure for the lower uniform cubic flux density (Fig. 2). Correspondingly greater uniformity of the overturn values at different stations reflects the uniformity of radiation distribution. Less variability between bacterial concentrations at

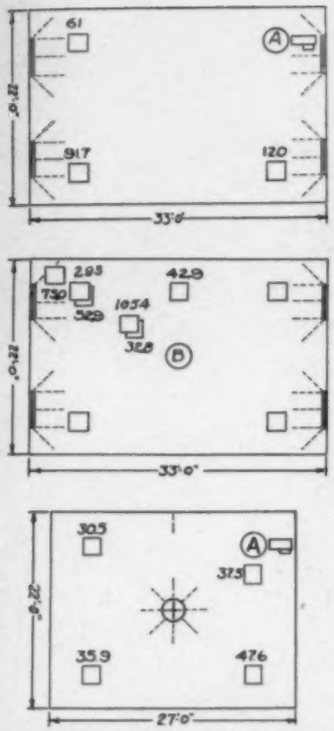


Fig. 2—Ventilation tests of schoolrooms at Germantown (central light fixture) and Swarthmore (two wall fixtures at each end) *A* and *B* represent positions of infector, or atomizer, squares represent positions of infectees, or air sampling centrifuges. Double squares represent pairs of machines, one sampling from upper level and one from bench level. Figures indicate lethes per hour between infector and infectee; air circulation counterclockwise.

upper and lower levels in the two installations also provides an empirical index of the ratio of circulation to disinfection rates.

Correlation of sanitary ventilation factors with irradiation factors, and foot lethes per occupant per minute with epidemiologic indices, will be required for codification. The purpose of this paper is to point out the nature of the data and the kind of instrument required for this purpose, and by presentation of actual data obtained under actual conditions of sanitary ventilation where hygienic values are clearly demonstrated, to emphasize the need of extending such methods to larger problems. Hygienic interpretation of sanitary ventilation is a problem for the sanitarian.

The sanitary significance of the actual values in the control of epidemic spread of contagion should not be misapplied to problems based on different premises, such as prevention of hospital cross-infection. Reduction of the hazard of breathing nuclei from droplets expelled by presumably well persons congregating with other presumably healthy members in an enclosed atmosphere is not the same as protecting the sick from breathing infected dust in a hospital ward, or preventing airborne infection of a surgical wound. An epidemic wave of contagion, due to prolonged but attenuated exposure in a form of infection highly vulnerable to irradiation may be checked by measures which could not guarantee complete prevention of sporadic cases. These are public health problems.

Air disinfection in surgical antisepsis has been fully demonstrated, but different methods of application and different techniques of evaluation apply to prevention of bacteria-laden dust or droplet nuclei settling into a wound. Design of ultraviolet barriers in the prevention of dust-borne hospital cross-infection in wards or nurseries requires different specifications. The flux density traversed by a dust particle passing from one cubicle to another must be far greater than would be sufficient for the sterilization of droplet nuclei. These are problems in hospital administration.

Modifications in technique, however, should not obscure the fact that ventilating engineers must seriously consider hygienic obligations imposed by any accepted theory that enclosed atmospheres constitute vehicles for the spread of infection. They but further multiply reasons for study of air-purification methods which promise to eliminate hygienic hazards from air supplies. With a country looking forward to a long struggle between massed populations, the problem of controlling epidemic spread of respiratory infections will become acute. Army concentrations within controllable atmospheres present special problems.

Acknowledgment

These collaborative studies between the School Boards of the Germantown Friends School and the Swarthmore Public Schools, the Hanovia Chemical and Manufacturing Co., the General Electric Co., and the Laboratories for the Study of Air-borne Infection, were made possible through the efforts of Dr. A. N. Richards, vice-president of the University of Pennsylvania, Dr. Joseph Stokes, Jr., professor of Pediatrics, University of Pennsylvania, Stanley Yarnall, principal of the Germantown Friends School, and Frank R. Morey, principal of the Swarthmore Public Schools. Bacteriologic data obtained by Richard Green, research associate, with the assistance of Miss Dorothy Wells, were also applied, and photometric measures were made by Arthur Hammerlund.

COLD ACCUMULATORS AND PANEL HEATING TOPICS AT MINNESOTA

January 5, 1942. The January meeting of the Minnesota Chapter was held at the Coffman Memorial Union on the Campus of the University of Minnesota, Minneapolis, with 25 members and guests present.

R. W. Evans and C. J. Otterholm explained the theory of design cooling, based on the total load instead of peak load conditions. To illustrate, they gave a problem in which they worked out the cooling load for an office in Minneapolis, basing their figures on total cooling load as well as peak load conditions.

Mr. Otterholm told of the history behind the development of the cold accumulator, the principle of its design and the advantages of using this equipment in various types of cooling jobs. He also told of their experience with its use during the past year where it has been in operation in various cities.

The members and guests were permitted to ask questions, and following a discussion the meeting adjourned.

December 1, 1941. The December meeting was held at the Center for Continuation Study on the University of Minnesota Campus, and was a joint dinner meeting with architects and others registered for the Institute in Building Methods and Materials. There were 65 members and guests in attendance.

The speakers for the evening were S. R. Lewis, consulting engineer, Chicago, and J. B. Fullman, Pittsburgh. Mr. Lewis discussed his experiences in the design of intercepter stations and various manufacturing plants used in connection with the Defense Program. Mr. Fullman told of the advantages of panel heating and described the methods of installation being used in different types of construction.

Officers Elected for 1942

E. O. Eastwood
Seattle, Wash.
President

A · S · H · V · E

M. F. Blankin
Philadelphia, Pa.
First Vice-President

S. H. Downs
Kalamazoo, Mich.
Second Vice-President

E. K. Campbell
Kansas City, Mo.
Treasurer

L. G. Miller
East Lansing, Mich.

A. J. Offner
New York, N. Y.

A. E. Stacey, Jr.
New York, N. Y.

B. M. Woods
Berkeley, Calif.

MEMBERS OF COUNCIL ELECTED FOR 1942—THREE-YEAR TERM

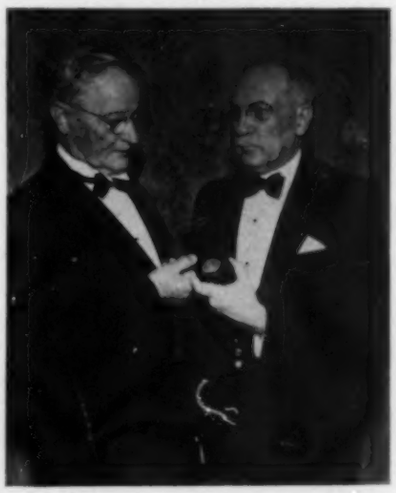
SOCIETY CONTRIBUTES TO VICTORY PROGRAM

BELIEVING that the national emergency requires an all-out effort from the engineering profession, the Society offered the full use of its facilities, including the specially trained personnel of its Research Laboratory, to the United States Government without expense, and subscribed $40,000 of its funds to defense bonds. This action was taken at the 48th Annual Meeting at the Bellevue-Stratford, Philadelphia, January 26-28, 1942, which attracted nearly 600 members and guests, the largest registration recorded at a meeting without an accompanying exposition.

With over 100 of its members already in the armed forces of the United States, and many in the allied armies, there was definite indication of the important work which Society members are doing and have already done in safeguarding the efficiency and health of men in camps, industrial workers, and civilians.

The technical sessions were heavily attended and many reports were received from the Society's Research Laboratory and from the cooperative projects carried on in Universities where work is sponsored by the Society's Committee on Research.

The final session was a joint meeting with the *National Warm Air Heating and Air Conditioning Association,* and the capacity of the Rose Room was taxed, due to the interest shown in the subjects under discussion. A stimulating address on Smoke Control for Cities was given by R. R. Tucker, Commissioner of Smoke Regulation for the City of St. Louis. He outlined the measures taken to remove the causes of the smoke nuisance, the educational program for the public, and pointed out that regulations covering hand-fired and mechanical fuel burning equipment are applied to domestic users, as well as industry and railroads. Figures compiled by the U. S. Weather Bureau indicated that smoke pollution in St. Louis had been reduced 72 per cent, and today residents of St. Louis enjoy practically the same hours of sunshine as residents in the surrounding country districts.



Philadelphia Inquirer Photo.
Dr. F. E. Giesecke Receives F. Paul Anderson Medal from Thornton Lewis

A feature of the annual banquet was the presentation of the F. Paul Anderson Medal to Dr. F. E. Giesecke, professor emeritus at A. and M. College of Texas, College Station, and past-president of the Society. The presentation was made by Thornton Lewis, Army Ordnance Department, who created the F. Paul Anderson Fund, and who

ASHVE Purchases Defense Bonds (*left to right*) Pres. W. L. Fleisher, Benj. Ludlow, State Dir., Defense Bond Sales, Percy C. Maderia, Jr., Pres., Land Title Bank and Trust Co., and Treas. M. F. Blankin

Edgar A. Kirby, Asst. Chief London Fire Force, Explains Incendiary Bomb Action to Pres. Fleisher and Treas. Blankin

served as president of the Society in 1929.

Dr. Frederick Ernest Giesecke received the F. Paul Anderson Gold Medal for distinguished scientific achievement.

The Committee on Award honored Dr. Giesecke for "his notable contributions to the advancement of heating based on his research work in the fields of heat transfer and hot water heating, and for his eminent services to the Society as a member and Officer."

Dr. Giesecke was born in Washington County, Texas in 1869 and, after receiving his preparatory education in the New Braunfels and San Antonio schools, he attended the Agricultural and Mechanical College of Texas, where he graduated in 1890. In 1906 and 1907 he studied abroad under Dr. H. Rietschel and subsequently he received degrees from the Massachusetts Institute of Technology and the University of Illinois.

Dr. Giesecke commenced his teaching career at A. & M. College of Texas in 1886, where he taught until 1912, after which he became Professor of Architecture at the University of Texas, Austin, as well as consulting architect, specializing in heating and ventilating. In 1927 he assumed the directorship of the Engineering Experiment Station at A. & M. College of Texas, a position he held until his retirement in 1939.

In 1940 he was President of the Society and has also held the offices of First Vice-President and Second Vice-President, as well as serving on innumerable important Technical Committees of the Society.

The F. Paul Anderson Medal Committee on Award for 1941 consisted of Prof. E. O. Eastwood, Seattle, Wash.; D. S. Boyden, Shirley, Mass.; W. T. Jones, Boston, Mass.; E. Holt Gurney, Toronto, Canada, and Prof. Perry West, Lexington, Ky. This award was created in 1931 by Thornton Lewis in honor of the late F. Paul Anderson, past president of the ASHVE, former director of its Research Laboratory and former Dean of the College of Engineering, University of Kentucky, Lexington, Ky.

W. L. Fleisher, New York, received the past-president's emblem from W. H. Driscoll, Syracuse, past-president of the Society and toastmaster A. J. Nesbitt, Philadelphia, introduced the newly elected officers: Prof. E. O. Eastwood, Seattle, Wash., *president;* Merrill F. Blankin, Philadelphia, *first vice-president;* S. H. Downs, Kalamazoo, Mich., *second vice-president;* and E. K. Campbell, Kansas City, Mo., *treasurer.*

The speaker of the evening was Stanley High, nationally known author, traveller, commentator and associate editor of Reader's Digest, whose subject was "What Are We Fighting for and Against?" In an inspiring address he said:

"The present war is not primarily an economic war but rather a conflict of ideas. The fight is not to save the British Empire but to prevent it from becoming a Japanese or a German empire. Revenge should not be the driving force in the conduct of the war. Enough hate may give us a peace that will make the war not worth winning. That is why the common man recognizes his stake in this war. He recognizes that unsatisfactory as his lot may be now, in a democracy he has a chance to improve it. He sees that that chance goes when Hitler comes."

The full report of the proceedings will appear in the March issue.

WESTERN N. Y. HEARS EFFORTS IN NATIONAL DEFENSE

January 12, 1942. The January meeting of the Western New York Chapter was held at the University Club, Buffalo, N. Y.

The March meeting which has been proposed as a joint meeting with the Toronto Chapter was discussed at some length and it was finally decided that Harry Schaefer and the Secretary would complete the necessary arrangements with H. R. Roth and V. J. Jenkinson of the Ontario Chapter.

President Heath then introduced the speaker of the evening, Lt. Col. J. F. Snyder, Commanding Officer in charge of the Western New York Medical Section of the U. S. Army, who gave a very interesting talk on *Raising Our Man Power Under the Selective Service Act.* As a fellow member of the Western New York Chapter, Colonel Snyder had an opportunity to meet many of his old friends and in the course of doing so gave a very interesting and constructive talk on his present efforts in National Defense.

ILLINOIS HEARS HOT WATER HEATING DISCUSSION

December 15, 1941. The Illinois Chapter meeting was held at the Top-of-the-Town Restaurant, Chicago, with Pres. I. E. Brooke presiding, and an attendance of 60 members and guests.

Following the regular order of business, R. E. Moore introduced the speaker of the evening, R. E. Hattis, consulting engineer and Past President of the Illinois Chapter, who spoke on the subject of hot water heating. Mr. Hattis discussed the use of hot water as a heat medium in heating and ventilating work, and pointed out that the prime requisite for comfort in all kinds of weather in all parts of the heated spaces is continuous delivery of heat in varying quantities as needed and where needed. He also brought out that in addition to low cost housing the use of forced circulation hot water is desirable in large projects in housing and in industrial, commercial and institutional uses.

In connection with the use of hot water for ventilating work, Mr. Hattis called attention to the use of parallel flow, and to the absolute uniformity of air leaving the coils when two or more pass coils are used. He made a few suggestions in connection with hot water systems emphasizing the desirability for proper venting and the necessity for adequate expansion tanks.

Mr. Hattis illustrated his talk with slides and concluded his remarks with the statement that hot water was a fine heating medium. A lively discussion followed in which several members and guests took part. The meeting adjourned at 9:45 p.m.

November 10, 1941. The November meeting was called to order by President Brooke with 67 members and guests present. C. M. Burnam, Jr., chairman of the meetings committee, announced that R. E. Hattis would be the guest at the next meeting, followed by the reading of a communication from the OPM in Washington expressing a desire for qualified men in the heating and ventilating field.

Following the reading of the minutes of the previous meeting, which were approved as read, the speaker of the evening, Edmund H.

Eitel, superintendent of analysts of the Priorities Field Service, of the OPM, was introduced. Mr. Eitel spoke on priorities in heating, ventilating and air conditioning work. In introducing the reasons for priorities, Mr. Eitel spoke briefly on the background of the war between the democracies and totalitarian states, and pointed out that everyone must sacrifice in consumption of civilian needs to divert production to defense. He then went into detail in discussing various specific priority orders of the OPM and explained something of the method of their use and how they function. The high-light of his talk was the question and answer period, which was interesting and instructive and thoroughly enjoyed. Following this worthwhile discussion the meeting adjourned at 10:05 p. m.

RADIO COMMENTATOR ADDRESSES MASSACHUSETTS

December 16, 1941. Approximately 25 members, guests and ladies of the Massachusetts chapter enjoyed a pleasant dinner and music by Jack Keyser and his accordion at the December gathering at the Hotel Kenmore.

Pres. J. W. Brinton opened the meeting with a *to the point* talk on the necessity of everyone cooperating in civilian defense work. D. J. Edwards very effectively introduced Ray Kierman, prominent news editor and radio commentator, whose talk on *Today's Headlines — How They Affect You and Me,* was very stimulating and created much favorable comment and discussion.

Tyler Stewart Rogers was announced as the speaker for the January meeting, and the meeting adjourned at 10:30 p. m.

STEAM HEATING AND DIRECT-FIRED HEATERS INTEREST PITTSBURGH

December 10, 1941. The regular meeting was called to order by Pres. E. C. Smyers in the Marine Room of the Hotel Roosevelt, Pittsburgh. The secretary read the minutes of the November meeting, which were approved as read.

Prof. C. M. Humphreys, chairman of the program committee, announced the plans for the January meeting, and in the absence of P. C. Strauch, chairman of the membership committee, President Smyers told of the progress on the membership roster.

President Smyers then introduced Messrs. Everetts and Foley, new chapter members and W. A. Evans, a guest from the Cleveland Chapter.

The guest speaker was H. F. Marshall, advertising and assistant sales manager, Warren Webster Co., who spoke on recent developments in steam heating. Mr. Marshall quickly traced the history of steam heating, during which he pointed out the problems in attaining a balanced distribution to all radiators. He then described how this balanced condition was accomplished by means of orifices and throttle valves. Mr. Marshall's talk was instructive as well as interesting, and was enjoyed by the 35 members and guests in attendance.

According to Secretary Rockwell's report, the meeting adjourned at 9:30 p. m. with a vote of thanks to Mr. Marshall.

November 10, 1941. Pres. E. C. Smyers called the November meeting of the Pittsburgh Chapter to order in the Gold Room of the Hotel Roosevelt, Pittsburgh. The minutes of the previous meeting were read and approved.

C. M. Humphreys, chairman of the program committee, announced the program for December, and the Secretary reported that the City Council is ready to take action on the proposed air conditioning code.

W. H. Reed introduced R. M. Rush as the guest speaker. Mr. Rush spoke on direct-fired heaters for factory heating, which type, he stated has been developed over a period of eight or nine years. He traced the history of its development, and then turned the meeting over for discussion and many questions were answered.

Upon motion of Professor Humphreys the 55 members and guests present extended a rising vote of thanks to Mr. Rush. The meeting adjourned at 9:20 p. m.

COIL DESIGN AND APPLICATION SUBJECT AT SOUTHERN CALIFORNIA

December 3, 1941. The regular meeting of the Southern California Chapter was held at Eaton's Chicken House on Wilshire Blvd., Los Angeles, with 43 members and guests present.

The meeting was called to order by Pres. A. J. Hess, and a short business session was held at which time J. B. Griffith explained that an educational display was to be provided at each meeting.

The business session was followed by a talk on the design and application of coils in air conditioning by Earl P. Wells, Los Angeles. The talk proved to be interesting and instructive. After a period of questions and answers the meeting was adjourned, according to Leo Hungerford, secretary of the Chapter.

DEFENSE COMMITTEE NAMED IN WASHINGTON, D. C.

December 10, 1941. The third regular meeting of the Washington, D. C. Chapter was held at the Dodge Hotel, with dinner at 6:30, followed by the usual business meeting in the Garden House under the leadership of Pres. F. A. Leser. There were 45 members and guests present.

Following the reading of the minutes which were approved as read, the treasurer submitted his report.

President Leser announced that the proposed Ladies' Night would have to be abandoned due to extenuating circumstances. He also announced the formation of a Defense Committee with Lt. Commdr. T. H. Urdahl as chairman and a Convention Committee with F. E. Spurney as chairman.

After announcements by J. H. Fogg on membership and H. J. Burns on the attendance, President Leser introduced the speaker of the evening, M. J. Wilson, Syracuse, N. Y. Mr. Wilson illustrated his talk which was on the subject of *Conduit System of Air Conditioning* and which was well received by his audience.

Following a period of discussion the meeting adjourned at 10:10 p. m., as reported by F. M. Thuney, secretary.

COUNCIL MEMBER ADDRESSES KANSAS CITY

December 8, 1941. After a buffet dinner, the meeting of the Kansas City Chapter was called to order by

Vice-Pres. M. M. Rivard, with 30 members present.

Following Treasurer Boyd's report, it was moved and seconded that Vice-President Rivard be appointed delegate to the Society's 48th Annual Meeting in Philadelphia, and W. A. Russell as alternate.

Mr. Rivard then announced that President Nottberg had previously appointed P. C. Leffel as chairman of the membership committee, and K. M. Stevens and Leon Mart as committee members.

L. P. Saunders, Lockport, N. Y., a member of the Society's Council, was then introduced as guest of the evening. Mr. Saunders gave an interesting talk on the design and application of heat exchangers for automotive, industrial, marine, and airplane use. His discussion was augmented with slides describing the details of the material presented. After his presentation an informal discussion was held and the speaker provided samples for the inspection of the members present, followed by adjournment.

WESTERN MICHIGAN HEARS FERDERBER

December 8, 1941. The December meeting of the Western Michigan Chapter was held at the Rowe Hotel, Grand Rapids, Mich., with 57 members and guests in attendance. The meeting was called to order by Pres. W. G. Schlichting, and the minutes of the previous meeting were read and approved.

President Schlichting then turned the meeting over to T. D. Stafford, who introduced the speaker of the evening, Dr. M. B. Ferderber, Pittsburgh, Pa., who gave a very interesting talk on the medical and engineering aspects of environmental conditions illustrating his remarks with slides showing the work being carried on in the treatment of various diseases.

Following a discussion President Schlichting called on F. C. Warren for a report on his plans for the January meeting. Mr. Stafford expressed his thanks to the members for their cooperation in making this meeting the success that it was, and on motion of O. D. Marshall and supported by H. J. Metzger the meeting was adjourned.

ATLANTA HOLDS ANNUAL MEETING

December 2, 1941. The 1941 annual meeting of the Atlanta Chapter was held at the East Lake Country Club.

The various committee chairmen reported for the year 1941 for their respective committees as follows: W. J. McKinney, *Finance;* C. L. Templin, *Membership;* M. M. Crout, *Programs;* F. L. Laseter, *Codes;* L. F. Kent, *Publicity;* T. T. Tucker, *Research.*

A copy of each of the committee reports is being made a part of the Chapter's minutes, and attention was called to the splendid work done by the membership committee under the leadership of Mr. Templin, which resulted in the Chapter's membership being increased from 28 members to 59 members.

Pres. S. W. Boyd then appointed Messrs. Kagey and Foss as Ballot Tellers for the election of officers for 1941. As a result of the balloting the following officers were elected for the year 1942:

President—L. F. Kent
Vice-President—A. H. Koch
Secretary—F. L. Laseter
Treasurer—M. M. Crout
Board of Governors—S. W. Boyd and C. B. Cole

Upon turning over the gavel to the incoming president, Mr. Boyd made the following recommendations to the Chapter:

1. Continuance of an active membership campaign and follow-up of prospects already approached.
2. Appointment of every member on one of the chapter committees.
3. Continuance of the policy set by the program committee of outlining the program one year in advance to assure a high quality of program.
4. Exertion of every effort to have the Society sponsorship of the attic fan research project for further work continued.

The incoming president, Mr. Kent, upon taking over the gavel, promised to try and keep up the fine example set by the previous president and then adjourned this portion of the meeting.

All members and guests then joined their wives and friends and the social part of the annual meeting took place. An excellent dinner was enjoyed, as well as dancing to the music of a 6-piece orchestra. There was a total of 80 members and guests in attendance.

According to Secretary Koch's report, this was one of the most successful annual meetings ever held by the Chapter, and it was declared the most successful meeting of its kind ever held by any engineering society in the City of Atlanta.

SMOKE ABATEMENT AND SOCIETY RESEARCH AT IOWA MEETING

November 17, 1941. The meeting was held at the Chamberlain Hotel, with Pres. Perry LaRue presiding. F. E. Triggs reported on the next program to be held at Waterloo, Ia.

President LaRue was appointed as delegate to the 48th Annual Meeting of the Society in Philadelphia, January 26-28, and Mr. Triggs was appointed as his alternate.

The first speaker, Cliff Millen, of the *Register and Tribune,* gave a very interesting account of the smoke abatement problems of St. Louis, explaining how the ordinance originated and how it is working up to the present. It was his opinion that 75 per cent of the people in St. Louis were in favor of the new ordinance. The chief ob-

Left to right: **Miss Evelyn Walters, F. L. Laseter, M. M. Crout, Mrs. G. R. Cary, L. F. Kent, S. W. Boyd, Mrs. S. W. Boyd, A. H. Koch, Mrs. A. H. Koch, W. J. McKinney and Mrs. W. J. McKinney**

jectors were those who could least afford to buy the better grades of fuel. He believed that St. Louis was the pioneer in the drastic smoke abatement program.

The second speaker, C. L. Campbell, from the Industrial Hygiene Department of the State Department of Health, presented other problems that may enter into the smoke abatement, such as pollution of the atmosphere from other gases. This presented an interesting discussion as to whether there were additional problems involved that should be considered in a smoke abatement ordinance beyond the reduction of objectionable smoke. Mr. Campbell pointed out that the main constituents of smoke were solids and gases. The solids were carbon, tar and ash, and these could be minimized with proper equipment.

Gilbert Perkins, Smoke Abatement Commissioner for the City of Des Moines, was the next speaker, and he advised what steps the City of Des Moines was taking in their smoke ordinance. He also advised that it was not as drastic as the St. Louis ordinance, and that they were concentrating on the larger plants. However, they are going step by step toward a satisfactory solution of the smoke problem in Des Moines.

October 13, 1941. President LaRue presided at the October meeting held at the Hotel Kirkwood, Des Moines, Ia., attended by 31 members and guests.

The minutes of the May 1941 meeting were read and approved. President LaRue outlined the year's tentative program, and asked for suggestions from the membership for programs, especially as to the type of meeting desired for the March meeting, at which it is expected to have a national speaker. Prof. R. A. Norman called attention to the new chapter committee recently appointed, and also asked for suggestions for the March meeting.

The program was presented by E. P. Heckel on the subject *Why Research.* He ably presented the research program of the Society from its beginning, and brought out many interesting facts that were new to the older as well as the younger members. Mr. Heckel pointed out the relation of research to the membership, and it was believed that a large, active membership would add to the research program of the Society. He called attention to the fact that outsiders, such as universities, schools, and manufacturers were contributing much more to the program of research than the membership. He stressed the fact that research should be the first interest of the members, and from the Society's inception, the importance of research has been emphasized, as its value was then unquestioned as it has been down to the present time.

Mr. Heckel presented a number of slides, illustrating the progress of research in the Society since the first research was established in 1919, calling attention to the fact that some of the original members of the Society at that time made special donations for research work which was in addition to their regular dues. He also called attention to the first GUIDE, which was published in 1922, and brought out the point that its data are accepted in this country and abroad as standard engineering practice. He pointed out that the Society was first organized in 1894, and the first local chapter was organized in 1903.

Mr. Heckel also outlined the work of Dr. R. W. Keeton, Department of Medicine, University of Illinois, on the subject of physiological reaction to environment. He gave a very interesting report on the work that is being carried on there at the present time.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 23 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by February 14, 1942, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BISHOP, MARION W., Sales Engr., American Blower Corp., Chicago, Ill. (*Advancement*)	John Howatt J. E. McClellan	S. R. Lewis J. H. Milliken
BULL, FREDERICK W., Engr., Newcomb & Boyd, Atlanta, Ga.	S. W. Boyd H. K. McCain	C. L. Templin C. B. Smoot
BYERS, ROBERT L., Engr., John Paul Jones, Cary & Millar, Cleveland, O.	J. P. Jones J. E. Wilhelm	Paul Gayman E. J. Sable
CLAPPERTON, ROBERT, Asst. Engr., Canadian Industries, Ltd., Mt. Royal, Que., Canada.	F. G. Ewens K. D. Leitch	A. J. Leitch J. P. Fitzsimons

CANDIDATES

HASTINGS, ADDISON B., Asst. Secy.-Sales Engr., Burnham Boiler Corp., Irvington, N. Y.

KEYES, MARCUS W., Federal Repr., Kimberly Clark Corp., Chevy Chase, Md.

KNAB, E. A., Htg. & Vtg. Contr., E. A. Knab Heating & Engineering Co., Milwaukee, Wis. (*Reinstatement*)

McWILLIAMS, JOSEPH W., Mech. Engr., Eastman Kodak Co., Rochester, N. Y.

MILLER, MAHLON S., Sales Engr., Iowa Public Service Co., Cherokee, Ia.

MINER, HARVEY H., Partner, Miner Supply Co., Red Bank, N. J. (*Advancement*)

MURRAY, H. G. S., Sales Engr., Canadian Comstock Co. Ltd., Toronto, Ont., Canada (*Advancement*)

RHEAULT, WALTER E., Research Engr., Young Radiator Co., Racine, Wis.

ST. LAURENT, GUY, Owner-Mgr., Hector Groulx Regd., Montreal, Canada.

SCHNEEBERG, FLOYD H., Engr., Twin City Furnace & Appliance Co., St. Paul, Minn.

SEELBACH, WILLIAM R., Pres., Equipment Sales, Inc., Buffalo, N. Y.

SOCKWELL, CHARLES, JR., Partner, Sockwell Co., Atlanta, Ga.

SOCKWELL, CHARLES, SR., Sr. Partner, Sockwell Co., Dunwoody, Ga.

SOCKWELL, TYRUS R., Partner, Sockwell Co., Atlanta, Ga.

STEMPEL, EDWARD H., Design Engr., Young Radiator Co., Racine, Wis.

SULLIVAN, THOMAS J., Student, Carnegie Institute of Technology, Pittsburgh, Pa.

USHER, WELLINGTON J., JR., Sales Engr., Arthur S. Leitch Co. Ltd., Toronto, Ont., Canada.

WHITEHURST, BERT W., Engr., Stone & Webster Engineering Co., Boston, Mass.

ZEIGLER, DONALD D., Student, Carnegie Institute of Technology, Pittsburgh, Pa.

REFERENCES

Proposers	*Seconders*
A. P. Weiss A. E. Bastedo	H. W. Fiedler O. O. Oaks
F. A. Leser J. H. Fogg	E. G. Brunner F. C. Hackett
J. A. Cutler W. A. Ouweneel	O. A. Trostel H. W. Schreiber
Richard Vidale H. L. Baker	R. P. Cook H. R. Patterson (*Non-member*)
C. E. Wiser E. H. Danielson	F. L. Adams R. A. Norman
A. H. Schroth K. L. Mytinger	I. E. Brex R. P. Strevell
Cyril Tasker H. R. Roth	W. C. Kelly V. J. Jenkinson
M. F. May	F. M. Young (*Non-member*)
F. H. E. Weil	W. A. Minkler
Leo Garneau F. A. Hamlet	G. L. Wiggs W. W. Hughes
B. W. Peterson H. M. Betts	R. W. Evans A. B. Algren
Herman Seelbach, Jr. D. J. Peterson	M. C. Beman B. C. Candee
C. L. Templin T. T. Tucker	L. F. Lawrence, Jr. A. H. Koch
C. L. Templin T. T. Tucker	L. F. Lawrence, Jr. A. H. Koch
C. L. Templin T. T. Tucker	L. F. Lawrence, Jr. A. H. Koch
F. H. E. Weil W. A. Minkler	F. M. Young (*Non-member*) J. J. Hilt (*Non-member*)
T. F. Rockwell C. M. Humphreys	P. A. Edwards J. F. Collins, Jr.
A. S. Leitch H. R. Roth	J. P. Fitzsimons A. M. Dion
H. G. Meinke W. W. Timmis	W. L. Fleisher C. E. Pabst
C. M. Humphreys T. F. Rockwell	P. A. Edwards J. F. Collins, Jr.

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BEACH, RALPH L., Branch Engr., York Ice Machinery Corp., Atlanta, Ga.

BENESCH, EDWARD J., Mech. Engr., Syska & Hennessy, New York, N. Y.

GREILING, WINFORD W., Mech. Engr., Austin Co., Detroit, Mich.

KIDD, CHARLES R., Htg. & Vtg. Engr., Defense Housing, Washington, D. C. (*Advancement*)

LIVELY, GEORGE P., Engr. (Naval Archt.), Navy Dept., Bureau of Ships, Washington, D. C.

LUDWIG, WILLIS D., Contract Engr., York Ice Machinery Corp., Atlanta, Ga.

SHIRE, A. C., Tech. Director, U. S. Housing Authority, Washington, D. C.

SMOOT, CHARLES B., Chief Engr., Carrier Corp., Atlanta, Ga.

ASSOCIATES

BANKS, JOHN B., Branch Mgr., Minneapolis-Honeywell Regulator Co., Portland, Ore. (*Reinstatement*)

BEHRER, WILLIAM H., JR., Htg. & Air Cond. Engr., Behrer Nason Co., Mineola, L. I., N. Y.

BENNETT, MERLE F., Chief Engr., First National Bank Bldg. Co., Detroit, Mich.

BRUCE, MARSHALL, Asst. Secy., George W. Akers Co., Detroit, Mich.

HIMSEL, STANLEY R., Mgr. (Htg. Dept.) L. L. Felker Co., Marshfield, Wis.

NORTH, SAMUEL L., Partner, North Bros., Atlanta, Ga.

OTTS, JOHN G., Sales Engr., York Ice Machinery Corp., Atlanta, Ga.

PINTER, JOSEPH L., Owner, Joseph L. Pinter & Co., Morristown, N. J.

SCHERGER, FRED J., Sales Engr., Mundet Cork Corp., Detroit, Mich.

WATSON, WILLIAM W., Mfrs. Agent., Detroit, Mich.

JUNIORS

CLARK, ALLAN M., Sales Engr., Canadian Blower & Forge Co. Ltd., Toronto, Ont., Canada.

KLINK, ERWIN J., Mech. Engr., Johnson Service Co., Detroit, Mich.

SIMMS, HERMAN, Draftsman, York Ice Machinery Corp., Atlanta, Ga.

SMITH, WILLIAM P., JR., Engr. & Estimator, The A. Z. Price Co. Inc., Charlotte, N. C.